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International

A TREATISE

ON

REFRIGERATING
AND
ICE-MAKING MACHINERY

PREPARED FOR STUDENTS OF
THE INTERNATIONAL CORRESPONDENCE SCHOOLS
SCRANTON, PA.

Volume II

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PNEUMATICS
HEAT
STEAM AND STEAM ENGINES
PRINCIPLES OF REFRIGERATION
REFRIGERATING AND ICE-MAKING MACHINERY

WITH PRACTICAL QUESTIONS AND EXAMPLES

First Edition

SCRANTON
THE COLLIERY ENGINEER CO.
1899

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PNEUMATICS.

PROPERTIES OF AIR AND GASES.

1039. **Pneumatics** is that branch of Mechanics which treats of the properties of gases.

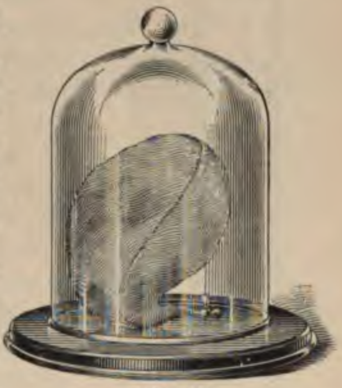

FIG. 188.

1040. The most striking feature concerning gases is that, *no matter how small the quantity may be, they will always fill the vessels which contain them.* If a bladder or football is partly filled with air and placed under a glass jar (called a **receiver**), from which the air has been exhausted, the bladder or football will immediately expand, as shown in Fig. 188. The force which a gas always exerts when confined in a limited space, is called **tension.** The word tension in this case means pressure, and is only used in this sense in reference to gases.

1041. As *water* is the most common type of fluids, so *air* is the most common type of gases. It was supposed by the ancients that air was imponderable, i. e., that it weighed nothing, and it was not until about the year 1650 that it was proven that air really had weight. A cubic inch of air, under ordinary conditions, weighs .31 grain, nearly. The ratio of the weight of air to water is about 1 : 774; that is, air is only $\frac{1}{774}$ as heavy as water. It can be shown experimentally that if a body be immersed in water, and weighs less than the volume of water displaced, the body will rise and extend partly out of the water. The same is true to a certain extent of air. If a vessel made of light material is filled with a gas lighter than air, so that the total weight of the vessel and gas is less than the weight of the volume of air which they displace, the vessel will rise. It is on this principle that balloons are made.

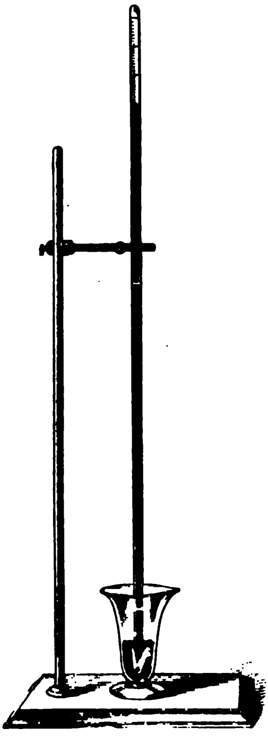

FIG. 189.

1042. Since air has weight, it is evident that the enormous quantity of air that constitutes the atmosphere must exert a considerable pressure upon the earth. This is easily proven by taking a long glass tube, closed at one end, and filling it with mercury. If the finger is placed over the open end, so as to keep the mercury from running out, and the tube is inverted and placed in a cup of mercury, as shown in Fig. 189, the mercury will fall, then rise, and after a few oscillations will come to rest at a height above the top of the mercury in the glass equal to about 30 inches. This height will always be the same under the same atmospheric conditions (allowance being made for the effects of capillary attraction). Now, if the atmosphere has weight, it must press upon the upper surface of the mercury in the glass with equal intensity upon every square unit, except upon that part of the surface occupied by the tube. According to a well-known law of hydromechanics, this pressure is transmitted in all directions. There being nothing in the tube, except the mercury, to counterbalance the upward pressure of the air, the mercury falls in the tube until it exerts a downward pressure on the upper surface of the mercury in the cup sufficiently great to counterbalance the upward pressure produced by the atmosphere. In order that there shall lhp

be equilibrium, the pressure of the air per unit of area on the upper surface of the mercury in the glass must equal the pressure (weight) exerted per unit of area by the mercury inside of the tube. Suppose that the area of the inside of the tube is one square inch ; then, since mercury is 13.6 times as heavy as water, the weight of the mercurial column is .03617 × 13.6 × 30 = 14.7574 pounds. The actual height of the mercury is a little less than 30 inches, and the actual weight of a cubic inch of distilled water is a little less than .03617 pound. When these considerations are taken into account, the average weight of the mercurial column at the level of the sea is 14.69 pounds, or, as it is usually expressed, 14.7 pounds. Since this weight, exerted upon 1 square inch of the liquid in the glass, just produced equilibrium, it is plain that the pressure of the outside air is 14.7 pounds upon every square inch of surface.

1043. Vacuum.—The space between the upper end of the tube and the upper surface of the mercury is called a **vacuum,** meaning that it is an entirely empty space, and does not contain any substance, solid, liquid, or gaseous. If there was a gas of some kind there, no matter how small the quantity might be, it would expand, filling the space, and its tension would cause the column of mercury to fall and become shorter, according to the amount of gas or air present. The space is then called a **partial vacuum.** If the mercury fell 1 inch, so that the column was only 29 inches high, we should say, in ordinary language, that there were *29 inches of vacuum.* If it fell 8 inches, we would say that there were 22 inches of vacuum ; if it fell 16 inches, we would say that there were 14 inches of vacuum, etc. Hence, when the vacuum gauge of a condensing engine shows 26 inches of vacuum, there is enough air in the condenser to produce a pressure of $\frac{30-26}{30} \times 14.7 = \frac{4}{30} \times 14.7 = 1.96$ pounds per square inch. In all cases where the mercurial column is used to measure a vacuum, the height of the column in inches gives the number of inches of vacuum. Thus, if the column were 5 inches high, or the vacuum gauge showed 5 inches, the vacuum would be 5 inches.

If the tube had been filled with water instead of mercury, the height of the column of water to balance the pressure of the atmosphere would have been 30 × 13.6 = 408 inches = 34 feet. This means that if a tube were filled with water, inverted and placed in a dish of water in a manner similar to the experiment made with the mercury, that the resulting height of the column of water would be 34 feet.

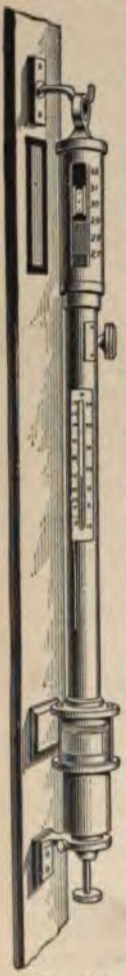

FIG. 190.

1044. The **barometer** is an instrument used for measuring the pressure of the atmosphere. There are two kinds in general use—the mercurial barometer and the aneroid barometer. The **mercurial barometer** is shown in Fig. 190. The principle is the same as in the case of the inverted tube shown in Fig. 189. The tube and cup at the bottom are protected by a brass or iron casing. At the top of the tube is a graduated scale which can be read to $\frac{1}{1000}$ of an inch, by means of a vernier. Attached to the casing is an accurate thermometer for determining the temperature of the outside air at the time the barometric observation is taken. This is necessary, since mercury expands when the temperature is increased, and contracts when the temperature falls; for this reason a standard temperature is assumed, and all barometer readings are reduced to this temperature. This standard temperature is usually taken at 32° F., at which temperature the height of the mercurial column is 30 inches. Another correction is made for the altitude of the place above sea level, and a third correction for the effects of capillary attraction.

1045. In Fig. 191 is shown a cut of an **aneroid barometer.** These instruments are made in various sizes, from the size of a large watch up to an 8 or 10 inch face. They consist of a cylindrical box of metal, with a top of thin, elastic, corrugated metal. The air is removed from the box. When the atmospheric pressure increases, the top is pressed inwards, and when it is diminished, the top is pressed outwards by its own elasticity, aided by a spring beneath. These movements of the cover are transmitted and multiplied by a combination of delicate levers which act upon an index hand and cause it to move either to the right or left over a graduated scale. These barometers are self-correcting (compensated) for variations in temperature. They are very portable, occupying but a small space, and are so delicate that they are said to show a difference in the atmospheric pressure when transferred from the table to the floor. They must be handled with care, as they are easily injured. The mercurial barometer is the standard.

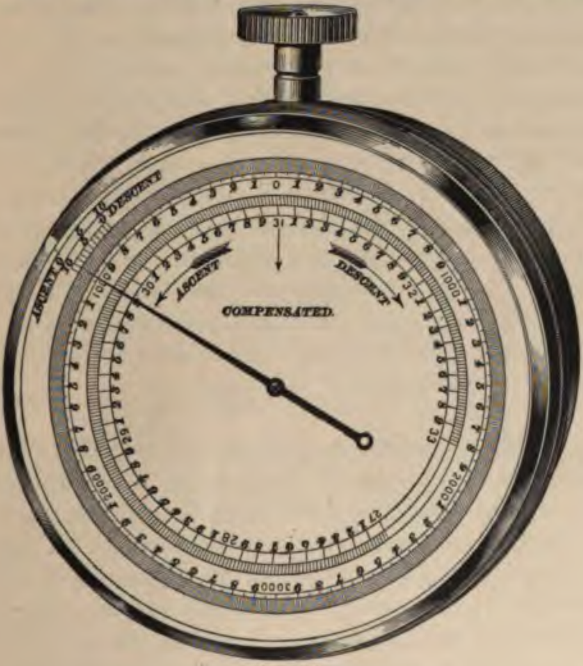

FIG. 191.

1046. With air, as with water, the lower we get, the greater the pressure, and the higher we get, the less the pressure. At the level of the sea, the height of the mercurial column is about 30 inches; at 5,000 feet above the sea, it is 24.7 inches; at 10,000 feet above the sea, it is 20.5 inches; at 15,000 feet above the sea, it is 16.9 inches; at 3 miles, it is 16.4 inches, and at 6 miles above the sea level, it is 8.9 inches.

The density also varies with the altitude; that is, a cubic foot of air at an elevation of 5,000 feet above the sea level will not weigh as much as a cubic foot at sea level. This is proved conclusively by the fact that at a height of 3½ miles the mercurial column measures but 15 inches, indicating that half the weight of the entire atmosphere is below that. It is known that the height of the earth's atmosphere is at least 50 miles; hence, the air just before reaching the limit must be in an exceedingly rarefied state. It is by means of barometers that great heights are measured. The aneroid barometer has the heights marked on the dial, so that it can be read directly. With the mercurial barometer, the heights must be calculated from the reading.

1047. The atmospheric pressure is everywhere present, and presses all objects in all directions with equal force. If a book is laid upon the table, the air presses upon it in every direction with an equal average force of 14.7 pounds per square inch. It would seem as though it would take considerable force to raise a book from the table, since, if the size of the book were 8 inches by 5 inches, the pressure upon it is $8 \times 5 \times 14.7 = 588$ pounds; but there is an equal pressure beneath the book to counteract the pressure on the top. It would now seem as though it would require a great force to open the book, since there are two pressures of 588 pounds each, acting in opposite directions, and tending to crush the book; so it would but for the fact that there is a layer of air between each leaf acting upwards and downwards with a pressure of 14.7 pounds per square inch. If two metal plates be made as perfectly smooth and flat as it is possible to get them, and the edge of one be laid upon the edge of the other, so that one may be slid upon the other, and the air thus excluded, it will take an immense force, compared with the weight of the plates, to separate them. This is because the full pressure of 14.7 pounds per square inch is then exerted upon each plate with no counteracting equal pressure between them.

If a piece of flat glass be laid upon a flat surface that has been previously moistened with water, it will require considerable force to separate them; this is because the water helps to fill up the pores in the flat surface and glass, and thus creates a partial vacuum between the glass and the surface, thereby reducing the counter pressure beneath the glass.

1048. Tension of Gases.—In Fig. 189 the space above the column of mercury was said to be a vacuum, and that if any gas or air was present, it would expand, its tension forcing the column of mercury downwards. If enough gas is admitted to cause the mercury to stand at 15 inches, the tension of the gas is evidently $\frac{14.7}{2} = 7.35$ pounds per square inch, since the pressure of the outside air of 14.7 pounds per square inch only balances 15 inches, instead of 30 inches, of mercury; that is, it balances only half as much as it would if there were no gas in the tube; therefore, the pressure (tension) of the gas in the tube is 7.35 pounds. If more gas is admitted until the top of the mercurial column is just level with the mercury in the cup, the gas in the tube has then a tension equal to the outside pressure of the atmosphere. Suppose that the bottom of the tube is fitted with a piston, and that the total length of the inside of the tube is 36 inches. If the piston be shoved upwards so that the space occupied by the gas is 18 inches long, instead of 36 inches, the temperature remaining the same as before, it will be found that the tension of the gas within the tube is 29.4 pounds per square inch. It will be noticed that the volume occupied by the gas is only half that in the tube before the piston was moved, while the pressure is twice as great, since 14.7 × 2 = 29.4 pounds. If the piston be shoved up, so that the space occupied by the gas is only 9 inches, instead of 18 inches, the temperature still remaining the same, the pressure will be found to be 58.8 pounds per square inch. The volume has again been reduced one-half, and the pressure increased 2 times, since 29.4 × 2 = 58.8 pounds. The space now occupied by the gas is 9 inches long, whereas, before the piston was moved it was 36 inches long; as the tube was assumed to be of uniform diameter throughout its length, the volume is now $\frac{9}{36} = \frac{1}{4}$ of its original volume, and its pressure is $\frac{58.8}{14.7} = 4$ times its original pressure. Moreover, if the temperature of the confined gas remains the same, the pressure and volume will always vary in a similar way. The law which states these effects is called *Mariotte's Law*, and is as follows:

1049. Mariotte's Law.—*The temperature remaining the same, the volume of a given quantity of gas varies inversely as the pressure.*

The meaning of this is: If the volume of the gas is diminished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc., of its former volume, the tension will be increased 2, 3, 5, etc., times, or if the outside pressure be increased 2, 3, 5, etc., times, the volume of the gas will be diminished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc., of its original volume, the temperature remaining constant. It also means that if a gas is under a certain pressure, and the pressure is diminished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{10}$, etc., of its original pressure, that the volume of the confined gas will be increased 2, 3, 10, etc., times—its tension decreasing at the same rate.

Suppose 3 cubic feet of air to be under a pressure of 60 pounds per square inch in a cylinder fitted with a movable piston, then, the product of the volume and pressure is 3 × 60 = 180. Let the volume be increased to 6 cubic feet, then the pressure will be 30 pounds per square inch, and 30 × 6 = 180, as before. Let the volume be increased to 24 cubic feet, it is then $\frac{24}{3} = 8$ times its original volume, and the pressure is $\frac{1}{8}$ of its original pressure, or $60 \times \frac{1}{8} = 7\frac{1}{2}$ pounds, and $24 \times 7\frac{1}{2} = 180$, as in the two preceding cases. It will now be noticed that if a gas be enclosed within a confined space, and allowed to expand without losing any heat, *the product of the pressure, and the corresponding volume for one position of the piston, is the same as for any other position of the piston.* If the piston were to compress the air, the same result would be obtained.

Let p = pressure for one position of the piston;
p_1 = pressure for any other position of the piston;
v = volume corresponding to the pressure p;
v_1 = volume corresponding to the pressure p_1.

$$\text{Then, } p\,v = p_1\,v_1. \qquad \textbf{(53.)}$$

1050. Knowing the volume and the pressure for any position of the piston, and the volume for any other position, the pressure may be calculated, or, if the pressure is known for any other position, the volume may be calculated.

EXAMPLE.—If 1.875 cubic feet of air be under a pressure of 72 pounds per square inch (*a*) what will be the pressure when the volume is increased to 2 cubic feet? (*b*) to 3 cubic feet? (*c*) to 9 cubic feet?

SOLUTION.—Solving formula **53,** for p_1, the unknown pressure,

(*a*) $p_1 = \frac{p\,v}{v_1} = \frac{72 \times 1.875}{2} = 67\frac{1}{2}$ lb. per sq. in. Ans.

(*b*) $p_1 = \frac{72 \times 1.875}{3} = 45$ lb. per sq. in. Ans.

(*c*) $p_1 = \frac{72 \times 1.875}{9} = 15$ lb. per sq. in. Ans.

EXAMPLE.—Ten cubic feet of air have a tension of 5.6 pounds per square inch; (*a*) what is the volume when the tension is 4 pounds? (*b*) 8 pounds? (*c*) 25 pounds? (*d*) 100 pounds?

SOLUTION.—Solving formula **53,** for v_1,

(*a*) $v_1 = \frac{p\,v}{p_1} = \frac{5.6 \times 10}{4} = 14$ cu. ft. Ans.

(*b*) $v_1 = \frac{5.6 \times 10}{8} = 7$ cu. ft. Ans.

(*c*) $v_1 = \frac{5.6 \times 10}{25} = 2.24$ cu. ft. Ans.

(*d*) $v_1 = \frac{5.6 \times 10}{100} = .56$ cu. ft. Ans.

pressure is the same as before, or 10 pounds per square inch. If more heat is applied until the temperature of the gas is 34° F., it will be found that the piston has again risen, and the volume again increased, while the pressure still remains the same. It will be found that for every increase of temperature there will be a corresponding increase of volume. The law which expresses this change, is called *Gay-Lussac's Law*, and is as follows:

1054. Gay-Lussac's Law.—*If the pressure remains constant, every increase of temperature of 1° F. produces in a given quantity of gas an expansion of $\frac{1}{492}$ of its volume at 32° F.*

If the pressure remains constant, it will also be found that every decrease of temperature of 1° F., will cause a decrease of $\frac{1}{492}$ of the volume at 32° F.

Let v = original volume of gas;
v_1 = final volume of gas;
t = temperature corresponding to volume v;
t_1 = temperature corresponding to volume v_1.

Then,
$$v_1 = v\left(\frac{460 + t_1}{460 + t}\right). \qquad (\mathbf{58.})$$

That is, *the volume of gas after heating (or cooling) equals the original volume, multiplied by 460, plus the final temperature, divided by 460, plus the original temperature.*

EXAMPLE.— 5 cubic feet of air at a temperature of 45°, are heated under constant pressure up to 177°; what is its volume?

SOLUTION.—Applying formula **58**,
$$v_1 = v\left(\frac{460 + t_1}{460 + t}\right) = 5\left(\frac{460 + 177}{460 + 45}\right) = 6.307 \text{ cu. ft.} \quad \text{Ans.}$$

1055. Suppose that a certain volume of gas is confined in a vessel so that it cannot expand; in other words, suppose that the piston of the cylinder before mentioned to be fastened so that it cannot move. Let a gauge be placed on the cylinder so that the tension of the confined gas can be registered. If the gas is heated, it will be found that for every increase of temperature of 1° F., there will be a corresponding increase of $\frac{1}{492}$ of the tension. That is, the volume remaining constant, the tension increases $\frac{1}{492}$ of the original tension for every degree rise of temperature.

Let p = the original tension;
t = the corresponding temperature;
p_1 = final tension;
t_1 = final temperature.

Then, $$p_1 = p\left(\frac{460 + t_1}{460 + t}\right). \qquad \textbf{(59.)}$$

That is, *if a certain quantity of gas be heated (or cooled) from $t°$ to $t_1°$, the volume remaining constant, the resulting tension p_1 will be equal to the original tension, multiplied by 460, plus the final temperature, divided by 460, plus the original temperature.*

Example.—If a certain quantity of air is heated under constant volume from 45° to 177°, what is the resulting tension, the original tension being 14.7 pounds per square inch?

Solution.—Applying formula **59**,

$$p_1 = p\left(\frac{460 + t_1}{460 + t}\right) = 14.7\left(\frac{460 + 177}{460 + 45}\right) = 18.542 \text{ lb. per sq. in. Ans.}$$

1056. According to the modern and now generally accepted theory of heat, the atoms and molecules of all bodies are in an incessant state of vibration. The vibratory movement in the liquids is faster than in the solids, and in the gases, faster than in either of the other two. Any increase of heat increases the vibrations, and a decrease of heat decreases them. From experiments and calculations based upon higher mathematics, it has been concluded that at 460° below zero, on the Fahrenheit scale, all these vibrations cease. This point is called the **absolute zero,** and all temperatures reckoned from this point are called the **absolute temperatures.** The point of absolute zero has never been reached, the lowest recorded temperature being about 393° F. below zero, but, nevertheless, it has a meaning, and is used in many formulas, being nearly always denoted by T. The ordinary temperatures are denoted by t. When the word temperature alone is used, the meaning is the same as ordinarily used, but when absolute temperature is specified, 460° F. must be added to the temperature.

The absolute temperature corresponding to 212° F. is 460 + 212 = 672° F. If the absolute temperature is given, the ordinary temperature may be found by subtracting 460 from the absolute temperature. Thus, the absolute temperature being 520° F., what is the temperature?

$$520° - 460° = 60°.$$

Let p = pressure in pounds per square inch;
V = volume of air in cubic feet;
T = absolute temperature;
W = weight in pounds.

Then, $pV = .37052\ T.$ **(60.)**

That is, *the pressure in pounds per square inch, multiplied by the volume of the air in cubic feet, equals .37052 times the absolute temperature corresponding to the pressure p and volume V.*

In this formula, the weight of the air is 1 pound.

EXAMPLE.—The pressure upon 9 cubic feet of air weighing 1 pound is 20 pounds per square inch; what is the temperature?

SOLUTION.—Applying formula **60**, $p\,V = .37052\ T$, or $20 \times 9 = .37052\ T$; hence, $T = \frac{180}{.37052} = 485.8°$, nearly. $485.8° - 460 = 25.8°$, the temperature. Ans.

EXAMPLE.—What is the volume of 1 pound of air whose temperature is 60° F. under a pressure of one atmosphere?

SOLUTION.—Applying formula **60**, $p\ V = .37052\ T$. Substituting, $14.7 \times V = .37052 \times (460 + 60) = .37052 \times 520$, or $V = \frac{.37052 \times 520}{14.7} = 13.107$ cubic feet. Ans.

1057. If the weight of the air be greater or less than 1 pound, the following formula must be used:

$pV = .37052\ W\ T.$ **(61.)**

That is, *the pressure in pounds per square inch, multiplied by the volume in cubic feet, equals .37052 times the weight in pounds multiplied by the absolute temperature.*

EXAMPLE.—3 cubic feet of air weighing .35 pound, are under a pressure of 48 pounds per square inch; what is the temperature of the air?

SOLUTION.—Applying formula **61**, $p\,V = .37052\ W\,T$. Substituting, $48 \times 3 = .37052 \times .35 \times T$, or $T = \frac{48 \times 3}{.37052 \times .35} = 1{,}110.4°$. Then, $1{,}110.4 - 460 = 650.4°$. Ans.

Example.—What is the weight of 1 cubic foot of air at a temperature of 32°, and under a pressure of one atmosphere?

Solution.—Applying formula **61**, $pV = .37052\,WT$. Substituting, $14.7 \times 1 = .37052 \times (460 + 32) \times W$, or

$$W = \frac{14.7}{.37052 \times 492} = .0806382 \text{ lb. Ans.}$$

If the pressure be taken as 14.69856 pounds per square inch, and the absolute zero as 459.4°, instead of 460° below zero, and if .370514 be used, instead of .37052, more exact values, the weight of 1 cubic foot would be $\frac{14.69856}{.370514 \times 491.4} = .08073$ lb., nearly.

Example.—What is the exact volume of 1 pound of air at a temperature of 32°, and at a pressure of one atmosphere? Take absolute zero at 459.4, and the pressure as 14.69856 pounds per square inch.

Solution.— $pV = .370514\,WT$, or $14.69856 \times V = .370514 \times 1 \times (459.4 + 32)$. $V = \frac{.370514 \times 491.4}{14.69856} = 12.387$ cu. ft. Ans.

1058. If in the formula $pV = .37052\,WT$, both sides of the equation be divided by T (which, of course, does not alter the equality), there results the expression $\frac{pV}{T} = .37052\,W$. Let p_1, V_1 and T_1 represent the pressure, volume and temperature of the same weight of air in another state; then, $p_1V_1 = .37052\,WT_1$. Dividing both sides by T_1, $\frac{p_1V_1}{T_1} = .37052\,W$. Therefore, since $\frac{pV}{T}$ and $\frac{p_1V_1}{T_1}$ are equal to the same thing (i. e., $.37052\,W$), they are equal to each other, and

$$\frac{pV}{T} = \frac{p_1V_1}{T_1} \qquad (62.)$$

This very important formula is the complete expression of Gay-Lussac's law, and is true for any of the so-called permanent gases. It was from this formula that formulas **58** and **59** were derived. Thus, let the pressure be constant; then, $p = p_1$, and $\frac{pV}{T} = \frac{pV_1}{T_1}$, or $V_1 = \frac{VT_1}{T} = V\left(\frac{460 + t_1}{460 + t}\right)$. Similarly, letting the volume be constant, $V = V_1$, and $\frac{pV}{T}$

$= \frac{p_1 V}{T_1}$, or $p_1 = \frac{p T_1}{T} = p\left(\frac{460 + t_1}{460 + t}\right)$. So, also, by letting the temperature be constant, $T = T_1$ and $\frac{pV}{T} = \frac{p_1 V_1}{T}$, or $pV = p_1 V_1$, which is the same as formula **53.**

1059. In formulas **53, 62, 63,** and **64,** it matters not with what units the pressures and volumes are measured, except that they must be the same throughout the same example, and the pressures must always be *absolute pressures.*

EXAMPLES FOR PRACTICE.

1. A vessel contains 25 cubic feet of gas at a pressure of 18 pounds per square inch; if 125 cubic feet of gas having the same pressure are forced into the vessel, what will be the resulting pressure? Ans. 108 lb. per sq. in.

2. A pound of air has a temperature of 126°, and a pressure of 1 atmosphere; what volume does it occupy? Ans. 14.77 cu. ft.

3. The volume of steam in the cylinder of a steam engine at cut-off is 1.35 cubic foot, and the pressure is 85 pounds per square inch; if the pressure at the end of the stroke is 25 pounds per square inch, what is the new volume? Ans. 4.59 cu. ft.

4. A certain quantity of air has a volume of 26.7 cubic feet, a pressure of 19.3 pounds per square inch, and a temperature of 42°; what is the weight? Ans. 2.77 lb.

5. A receiver contains 180 cubic feet of gas at a pressure of 20 pounds per square inch; if a vessel holding 12 cubic feet, to be filled from the receiver until its pressure is 20 pounds per square inch, what will be the pressure in the receiver? Ans. 18⅔ lb. per sq. in.

6. 10 cubic feet of air having a pressure of 22 pounds per square inch, and a temperature of 75°, are heated until the temperature is 300°; the volume remaining the same, what is the new pressure? Ans. 31.25 lb. per sq. in.

7. If a spherical shell whose outside diameter is 18 inches, has a part of the air within it removed until the pressure is 5 pounds per square inch, what is the total pressure due to the atmosphere tending to crush the shell? Ans. 9,873.42 lb.

THE MIXING OF GASES.

1060. If two liquids which do not act chemically upon each other are mixed together and allowed to stand, it will be found that after a time the two liquids have separated, and that the heavier has fallen to the bottom. If two equal vessels, containing gases of different densities, be put in communication with each other, they will be found to have mixed in equal proportions after a short time. If one vessel be higher than the other, and the heavier gas be in the lower vessel, the same result will occur. The greater the difference of the densities of the two gases, the quicker they will mix. It is assumed that no chemical action takes place between the two gases. When the two gases have the same temperature and pressure, the pressure of the mixture will be the same; this is evident, since the total volume has not been changed, and unless the volume or temperature changes, the pressure cannot change. This property of the mixing of gases is a very valuable one, since, if they acted like liquids, carbonic acid gas (the result of combustion), which is $2\frac{1}{2}$ times as heavy as air, would remain next to the earth, instead of dispersing into the atmosphere, the result being that no animal life could exist.

1061. Mixtures of Equal Volumes of Gases Having Unequal Pressures.—*If two gases having equal volumes and temperatures, but different pressures, be mixed in a vessel whose volume equals one of the equal volumes of the gas, the pressure of the mixture will be equal to the sum of the two pressures, provided that the temperature remains the same as before.*

EXAMPLE.—Two vessels containing 3 cubic feet of gas, each at a temperature of 60°, and subjected to pressures of 40 pounds and 25 pounds per square inch, respectively, are placed in communication with each other, and all the gas is compressed into one vessel. If the temperature of the mixture is also 60°, what is the pressure?

SOLUTION.—According to the rule just given, the pressure will be $40 + 25 = 65$ pounds per square inch. This may be proven by applications of Mariotte's law; thus, compress the gas whose pressure is 25 pounds per square inch until its pressure is 40 pounds; its volume may be found thus: $p v = p_1 v_1$, or $25 \times 3 = 40 \times v$; whence, $v = 1.875$ cubic feet. Let communication be established between the two vessels, the pressure will evidently be 40 pounds and the total volume $3 + 1.875 = 4.875$ cubic feet. If this be compressed until the volume is 3 cubic feet, the temperature remaining at 60° throughout the whole operation, the final pressure may be found by formula **53**, $p v = p_1 v_1$.

Thus, $40 \times 4.875 = p_1 \times 3$, and $p_1 = \frac{40 \times 4.875}{3} = 65$ pounds per square inch, as before.

1062. Mixture of Two Gases Having Unequal Volumes and Pressures.

Let v and p be the volume and pressure, respectively, of one of the gases.

Let v_1 and p_1 be the volume and pressure, respectively, of the other gas.

Let V and P be the volume and pressure, respectively, of the mixture. Then, if the temperature remains the same,

$$V P = v p + v_1 p_1. \qquad \textbf{(63.)}$$

That is, *if the temperature is constant, the volume after mixture, multiplied by the resulting pressure, equals the volume of one gas before mixture multiplied by its pressure, plus the volume of the other gas multiplied by its pressure.*

EXAMPLE.—Two gases of the same temperature, having volumes of 7 cubic feet and 4½ cubic feet, and whose pressures are 27 pounds and 18 pounds per square inch, respectively, are mixed together in a vessel whose volume is 10 cubic feet. The temperature of the two gases and of the mixture being 60° F., what is the resulting pressure?

SOLUTION.—Applying formula **63**, $P V = p v + p_1 v_1$, or $P \times 10 = 27 \times 7 + 4\frac{1}{2} \times 18$. Hence, $P = \frac{189 + 81}{10} = 27$ lb. per sq. in. Ans.

1063. Mixture of Two Volumes of Air Having Unequal Pressures, Volumes, and Temperatures.

If a body of air having a temperature t_1, a pressure p_1, and a volume v_1 be mixed with another volume of air having a temperature t_2, a pressure p_2, and a volume v_2, to form a volume V having a pressure P and a temperature t, then, either the new temperature t, the new volume V, or the new pressure P may be found, if the other two quantities are known, by the following formula, in which T_1, T_2, and T are the absolute temperatures corresponding to t_1, t_2, and t:

$$P V = \left[\frac{p_1 v_1}{T_1} + \frac{p_2 v_2}{T_2}\right] T. \qquad \textbf{(64.)}$$

EXAMPLE.—Five cubic feet of air having a tension of 30 pounds per square inch, and a temperature of 80° F., are required to be compressed together with 11 cubic feet of air having a tension of 21 pounds per square inch, and a temperature of 45° F., in a vessel whose cubical contents are 8 cubic feet. The new pressure is required to be 45 pounds per square inch. What is the temperature of the mixture?

SOLUTION.—Substituting in formula **64**,

$45 \times 8 = \left[\frac{30 \times 5}{540} + \frac{21 \times 11}{505}\right] \times T$, or $360 = .7352\ T$. Hence, $T = \frac{360}{.7352} = 489.66°$, nearly, and $t = 29.66°$. Ans.

EXAMPLES FOR PRACTICE.

1. Two vessels contain air at pressures of 60 and 83 pounds per square inch. The volume of each vessel is 8.47 cubic feet. If all of the air in both vessels is removed to another vessel, and the new pressure is 100 pounds per square inch, what is the volume of the vessel, the temperature being the same throughout?
Ans. 12.11 cu. ft.

2. A vessel contains 11.83 cubic feet of air at a pressure of 33.3 pounds per square inch. It is desired to increase the pressure to 40 pounds per square inch by supplying air from a second vessel which contains 19.6 cubic feet of air at a pressure of 60 pounds per square inch. What will be the pressure in the second vessel after the pressure in the first has been raised to 40 pounds per square inch?
Ans. 55.96 lb. per sq. in.

3. If 4.8 cubic feet of air having a tension of 52 pounds per square inch and a temperature of 170° are mixed with 13 cubic feet having a tension of 78 pounds per square inch and a temperature of 265°, what must be the volume of the vessel containing the mixture in order that the tension of the mixture may be 30 pounds per square inch and the temperature 80°? Ans. 32.31 cu. ft.

PNEUMATIC MACHINES.

THE AIR PUMP.

1064. The **air pump** is an instrument for removing air from an enclosed space. A section of the principal parts is shown in Fig. 192, and the complete instrument in Fig. 193. The closed vessel *R* is called the **receiver**, and the space which it encloses is that from which it is desired to remove the air. The receiver is usually made of glass, and the edges are ground so as to be perfectly air-tight. When made in the form shown, it is called a **bell jar receiver.**

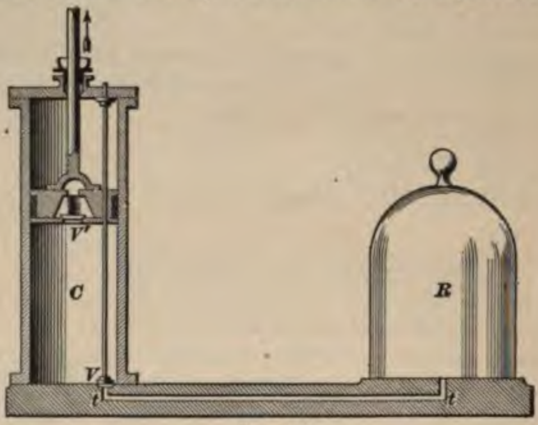

FIG. 192.

The receiver rests upon a horizontal plate in the center of which is an opening communicating with the pump cylinder C by means of a bent tube t. The pump piston fits the cylinder accurately, and has a valve V' opening upwards. At the junction of the tube with the cylinder is another valve V also opening upwards. When the piston is raised the valve V' closes, and, since no air can get into the cylinder from above, the piston leaves a vacuum behind it. The pressure on top of V being now removed, the tension of the air in the receiver R causes V to rise; the air in the receiver then expands and occupies the

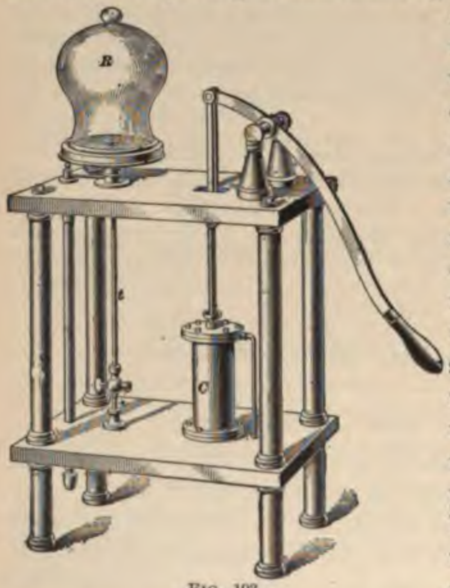

FIG. 193.

space displaced by the piston, the space in the tube *t* and in the receiver *R*. The piston is now pushed down, the valve *V* closes, the valve *V'* opens, and the air in *C* escapes. The lower valve *V* is sometimes supported, as shown in Fig. 192, by a metal rod passing through the piston and fitting it somewhat tightly. When the piston is raised or lowered, this rod moves with it. A button near the upper end of the rod confines its motion to within very narrow limits, the piston sliding upon the rod during the greater part of the journey.

1065. Degrees and Limits of Exhaustion.—Suppose that the volume of *R* and *t* together is four times that of *C*, and that there are, say, 200 grains of air in *R* and *t*, and 50 grains in *C*, when the piston is at the top of the cylinder. At the end of the first stroke, when the piston is again at the top, 50 grains of air in the cylinder *C* will have been removed, and the 200 grains in *R* and *t* will occupy the spaces *R*, *t*, and *C*. The ratio between the sum of the spaces *R* and *t* and the total space $R+t+C$ is $\frac{4}{5}$; hence, $200 \times \frac{4}{5} = 160$ grains = the weight of air in *R* and *t* after the first stroke. After the second stroke, the weight of the air in *R* and *t* would be $(200 \times \frac{4}{5} \times \frac{4}{5} = 200 \times (\frac{4}{5})^2 = 200 \times \frac{16}{25} = 128$ grains. At the end of the third stroke, the weight would be $[200 \times (\frac{4}{5})^2] \times \frac{4}{5} = 200 \times (\frac{4}{5})^3 = 200 \times \frac{64}{125} = 102.4$ grains. At the end of *n* strokes, the weight would be $200 \times (\frac{4}{5})^n$. It is evident that *it is impossible to remove all of the air that is contained in R and t by this method.* It requires an exceedingly good air pump to reduce the tension of the air in *R* to $\frac{1}{50}$ of an inch of mercury. When the air has become so rarefied as this, the valve *V'* will not lift, and, consequently, no more air can be exhausted.

1066. Sprengel's Air Pump.—In Fig. 194, *c d* is a glass tube longer than 30 inches, open at both ends, and connected by means of India rubber tubing with a funnel *A* filled with mercury and supported by a stand. Mercury is allowed to fall into this tube at a rate regulated by a clamp at *c*. The lower end of the tube *c d* fits in the flask *B*, which has a spout at the side a little higher than the lower end of *c d*; the upper part has a branch at *x* to which a receiver *R* can be tightly fixed. When the clamp at *c* is opened the first portions of the mercury which run out close the tube and prevent air from entering from below. These drops of mercury act like little pistons, carrying the air in front of them and forcing it out through the bottom of the tube. The air in *R* expands to fill the tube every time that a drop of mercury falls, thus creating a partial vacuum in *R*, which becomes more nearly complete as the process goes on. The escaping mercury falls into the dish *H*, from which it can be poured back into the funnel from time to time. As the exhaustion from *R* goes on, the mercury rises in the tube *c d* until, when the exhaustion is complete, it forms a continuous column 30 inches high; in other words, it is a barometer, whose Torricellian vacuum is the receiver *R*. This instrument necessarily requires a great deal of time for its operation, but the results are very complete, a vacuum of $\frac{1}{43000}$ of an inch of mercury being sometimes obtained. By use of chemicals in addition to the above, a vacuum of $\frac{1}{650000}$ of an inch of mercury has been obtained.

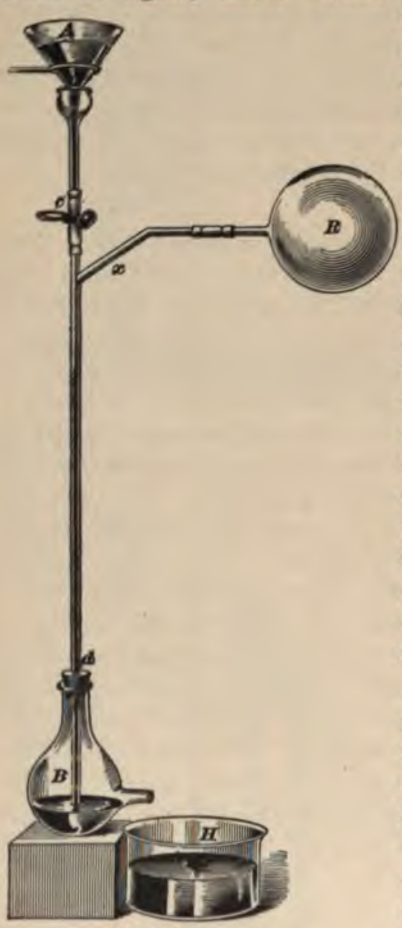

FIG. 194.

1067. NOTE.—A theoretically perfect vacuum is sometimes called a **Torricellian vacuum.**

1068. Magdeburg Hemispheres.—By means of the two hemispheres shown in Fig. 195, it can be proven that the atmosphere presses upon a body equally in all directions. They were invented by Otto Von Guericke, of Magdeburg, and are called the **Magdeburg hemispheres.** One of the hemispheres is provided with a stop-cock, by which it can be screwed on to an air pump. The edges fit accurately and are well greased, so as to be air-tight. As long as the hemispheres contain air, they can be separated with ease; but when the air in the interior is pumped out by means of an air pump, they can be separated only with great difficulty. The force required to separate them will be equal to the area of the largest circle of the hemisphere (projected area) in square inches, multiplied by 14.7 pounds.

This force will be the same in whatever position the hemisphere may be held, thus proving that the pressure of air upon it is the same in all directions.

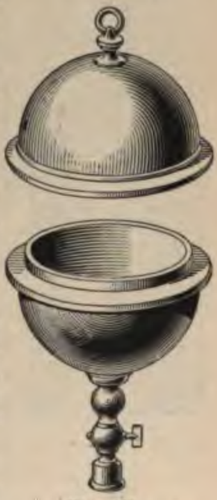

FIG. 195.

1069. The Weight Lifter.—The pressure of the atmosphere is very clearly shown by means of an apparatus like that illustrated in Fig. 196. Here, a cylinder fitted with a piston is held in suspension by a chain. At the top of the cylinder is a plug *A*, which can be taken out. This plug is removed, the piston pushed up (the force necessary being equal to the weight of the piston and rod *B*) until it touches the cylinder head. The plug is then screwed in, and the piston will remain at the top until a weight has been hung on the rod equal to the area of the piston, multiplied by 14.7 pounds, less the weight of the piston and rod. If a force was applied to the rod sufficiently great to force the piston downwards, it would raise any weight less than the above to the top of the cylinder. Suppose the weight to be removed, and the piston to be supported, say midway of the length of the cylinder. Let the plug be removed and air admitted above the piston, then screw the plug back into its place; if the piston be shoved upwards, the farther up it goes, the greater will be the force necessary to push it, on account of the compression of the air. If the piston is of large diameter, it will also require a great force to pull it out of the cylinder, as a little consideration will show. For example, let the diameter of the piston be 20 inches, the length of the cylinder 36 inches, plus the thickness of the piston, and the weight of the piston and rod 100 pounds. If the piston is in the middle of the cylinder, there will be 18 inches of space above it, and 18 inches of space below it. The area of the piston is $20^2 \times .7854 = 314.16$ square inches, and the atmospheric pressure upon it is $314.16 \times 14.7 = 4,618$ pounds, nearly. In order to shove the piston upwards 9 inches, the pressure upon it must be twice as great, or 9,236 pounds, and to this must be added the weight of the piston and rod, or $9,236 + 100 = 9,336$ pounds. The force necessary to cause the piston to move upwards 9 inches would then be $9,336 - 4,618 = 4,718$ pounds. Now, suppose the piston to be moved downwards until it is just on the point of being pulled out of the cylinder. The volume above it will then be twice as great as before, and the pressure one-half as great, or $4,618 \div 2 = 2,309$ pounds. The total upward pressure will be the pressure of the atmosphere less the weight of the piston and rod, or $4,618 - 100 = 4,518$ pounds, and the force necessary to pull it downwards to this point will be $4,518 - 2,309 = 2,209$ pounds.

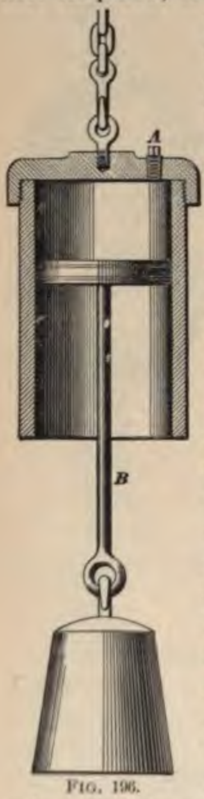

FIG. 196.

1070. The Baroscope.—The buoyant effect of air is very clearly shown by means of an instrument called the **baroscope,** shown in Fig. 197. It consists of a scale beam, from one extremity of which is suspended a small weight, and from the other a hollow copper sphere. In air they exactly balance each other; but when placed under the receiver of an air pump and the air exhausted, the sphere sinks, showing that it is really heavier than the small weight. Before the air is exhausted, each body is buoyed up by the weight of the air it displaces, and since the sphere displaces the most air, it loses more weight by reason of this displacement than the small weight. Suppose that the volume of the sphere exceeds that of the weight by 10 cubic inches; the weight of this volume of air is 3.1 grains. If this weight be added to the small weight, it will overbalance the sphere in air, but will exactly balance it in a vacuum.

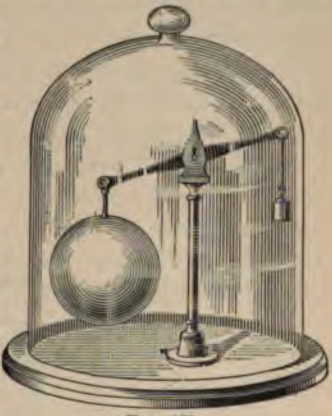

FIG. 197.

AIR COMPRESSORS.

1071. For many purposes compressed air is preferable to steam or other gas for use as a motive power. In such cases **air compressors** are used to compress the air. These are made in many forms, but the most common one is to place a cylinder, called the *air cylinder*, in front of the cross-head of a steam engine, so that the piston of the air cylinder can be driven by attaching its piston rod to the cross-head, in a manner similar to a steam pump. A cross-section of the air cylinder of a compressor of this kind is shown in Fig. 198, in which A is the piston and B is the piston rod, driven by the cross-head of a steam engine not shown in the figure. Both ends of the lower half of the cylinder are fitted with inlet valves D and D', which allow the air to enter the cylinder, and both ends of the upper half are fitted with discharge valves *F* and *F'*, which allow the air to escape from the cylinder after it has been compressed to the required pressure.

Suppose the piston *A* to be moving in the direction of the arrow; then the inlet valves *D* in the left-hand end of the cylinder from which the piston is moving will be forced inwards by the pressure of the atmosphere, which overcomes the resistance of the light spring *C*, thus allowing the air to flow in and fill the cylinder. On the other side of the piston, the air is being compressed, and, consequently, it acts with the springs *S* to force the inlet valves *D'* in the right-hand end of the cylinder to their seats. In the right-hand end of the cylinder, the discharge valves *F'* are opened when the pressure of the air in the cylinder is great enough to overcome the resistance of the light springs *E'*

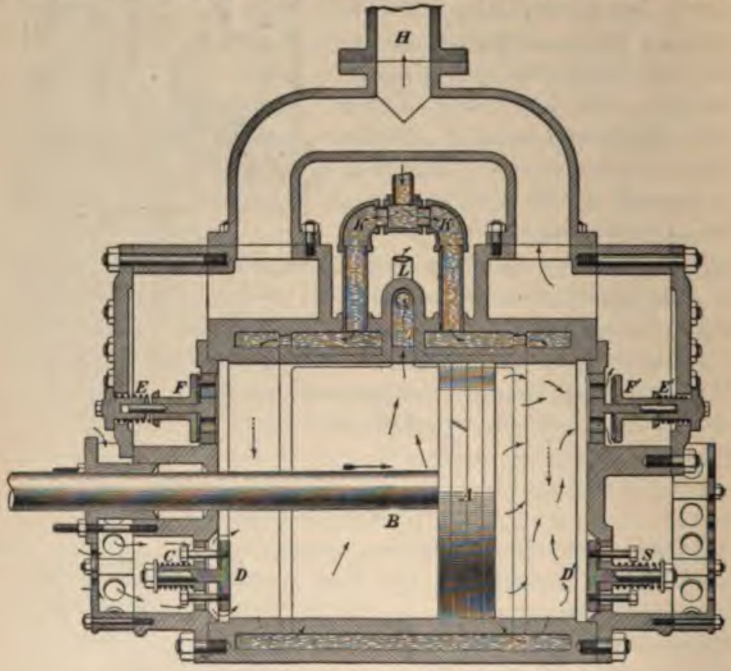

FIG. 198.

and the tension of the air in the passages leading to the discharge pipe *H*, and the discharge valves *F* are pressed against their seats by the springs *E* and the tension of the air in the passages. Suppose it is desired to compress the air to 59 pounds per square inch, and we wish to find at what point of the stroke the discharge valves will open. Now, 59 pounds per square inch equals a pressure of 4 atmospheres, very nearly; hence, when the pressure in the cylinder becomes great enough to force air out through the discharge valves, the volume must be one-quarter of the volume at atmospheric pressure, or the valves will open when the piston has traveled three-quarters of its stroke, provided the air be compressed at constant temperature.

The air, after being discharged from the cylinder, passes out through the delivery pipe *H*, and from thence is conveyed to its destination. It was shown in the early part of this paper that when air or any other gas was compressed its temperature was increased. For high pressures this increase of temperature becomes a serious consideration, for two reasons: 1st. When the air is discharged at a high temperature, the pressure falls considerably when it has cooled down to its normal temperature, and this represents a serious loss in the economical working of the machine. 2d. The alternate heating and cooling of the compressor cylinder by the hot and cold air is very destructive to it, and increases the wear to a great extent. To prevent the air from heating, cooling devices are resorted to, the most common one being the so-called **water jacket.** This is effected in the following manner: The cylinder walls are hollow, as shown in the cut; the cold water enters this hollow space in the cylinder wall through the pipe *K K*, and flows around the cylinder, finally passing out through the discharge pipe *L*. The water tends to keep the cylinder walls cold, and these cool the air as it is compressed.

1072. The Cartesian Diver.—The instrument shown in Fig. 199, called the **cartesian diver,** illustrates the elasticity of air and the transference of pressure in all directions in water. It consists of a glass jar filled with water, having a rubber bulb at the top filled with air. The image in the jar is made of glass and is hollow, the weight being less than an equal volume of water, so that it will float at the top of the jar. The tail of the image has a hole in it, the water being prevented from getting inside of the image by the tension of the air within it. If the bulb be squeezed, the air in it will be forced out, creating a pressure upon the water which, being transferred in all directions, causes the water to flow into the tail of the image, compressing the air inside, and thus causing it to fall to the bottom of the jar. When the bulb is released, the air flows back into it; the pressure upon the water is removed, the air within the image expands; the image, again becoming lighter than water, rises to the top of the jar.

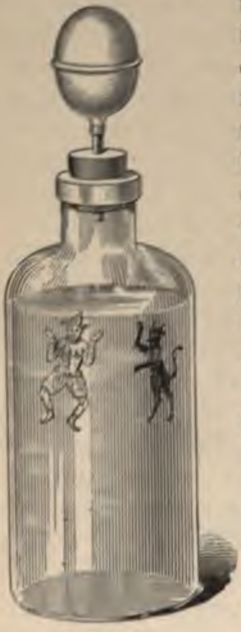

FIG. 199.

1073. Hero's Fountain.

Hero's fountain derives its name from its inventor, Hero, who lived at Alexandria 120 B. C. It is shown in Fig. 200. It depends for its operation upon the elastic properties of air. It consists of a brass dish *A*, and two glass globes *B* and *C*. The dish communicates with

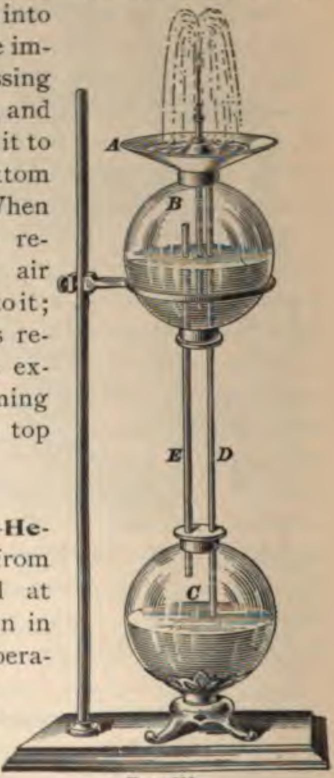

FIG. 200.

the lower part of the globe *C* by a long tube *D*, and another tube *E* connects the two globes. A third tube passes through the dish *A* to the lower part of the globe *B*. This last tube being taken out, the globe *B* is partially filled with water; the tube is then replaced and water is poured into the dish. The water flows through the tube *D* into the lower globe, and expels the air, which is forced into the upper globe. The air thus compressed acts upon the water and makes it jet out through the shortest tube, as represented in the figure. Were it not for the resistance of the atmosphere and friction, the water would rise to a height above the water in the dish equal to the difference of the level of the water in the two globes.

THE SIPHON.

1074. The action of the **siphon** illustrates the effect of atmospheric pressure. It is simply a bent tube of unequal branches, open at both ends, and is used to convey a liquid from a higher point to a lower, over an intermediate point higher than either. In Fig. 201, *A* and *B* are two vessels, *B* being lower than *A*, and *A C B* is the bent tube or siphon. Suppose this tube to be filled with water and placed in the vessels, as shown, with the short branch *A C* in the vessel *A*. The water will flow from the vessel *A* into *B*, so long as the level of the water in *B* is below the level of the water in *A*, and the level of the water in *A* is above the lower end of the tube *A C*. The atmospheric pressure upon the surfaces of *A* and *B* tends to force the water up the tubes *A C* and *B C*. When the siphon is filled with water, each of these pressures is counteracted in part by

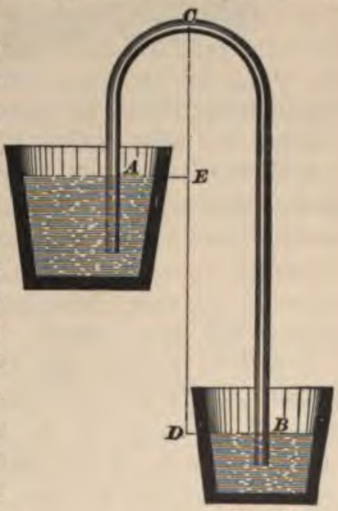

FIG. 201.

the pressure of the water in that branch of the siphon which is immersed in the water upon which the pressure is exerted. The atmospheric pressure opposed to the weight of the longer column of water will, therefore, be more resisted than that opposed to the weight of the shorter column; consequently, the pressure exerted upon the shorter column will be greater than that upon the longer column, and this excess pressure will produce motion.

Let A = the area of the tube in square inches.

$h = D\,C$ = the vertical distance in inches between the surface of the water in B and the highest point of the center line of the tube.

$h_1 = E\,C$ = the distance in inches between the surface of the water in A and the highest point of the center line of the tube.

The weight of the water in the short column is $.03617\,A\,h_1$, and the resultant atmospheric pressure, tending to force the water up the short column, is $14.7 \times A - .03617\,A\,h_1$. The weight of the water in the long column is $.03617\,A\,h$, and the resultant atmospheric pressure, tending to force the water up the long column, is $14.7\,A - .03617\,A\,h$. The difference between these two is $(14.7\,A - .03617\,A\,h_1) - (14.7\,A - .03617\,A\,h) = .03617\,A(h - h_1)$. But $h - h_1 = E\,D$ = the difference between the levels of the water in the two vessels. To find the discharge from a siphon, use the difference $h - h_1$, reduced to feet, as the head, and the total length of the siphon between the two water levels, as the length of the pipe; the discharge may then be calculated by formula **50,** Art. **1032.**

It will be noticed that the short column must not be higher than 34 feet for water, or the siphon will not work, since the pressure of the atmosphere will not support a column of water that is higher than 34 feet; 28 feet is considered to be the greatest height for which a siphon will work well.

1075. Intermittent Springs.—Sometimes a spring is observed to flow for a time and then cease; then, after an interval, to flow again for a time. The generally accepted explanation of this is that there is an underground reservoir fed with water through fissures in the earth, as shown in Fig. 202. The outlet for the water is shaped like a siphon, as shown. When the water in the reservoir reaches the same height as the highest point of outlet, it flows out until the level of the water in the reservoir falls below the mouth of the siphon, the water flowing out of the reservoir faster than it is supplied to it. This flow then ceases until the water in the reservoir has again reached the level of the highest point of the siphon.

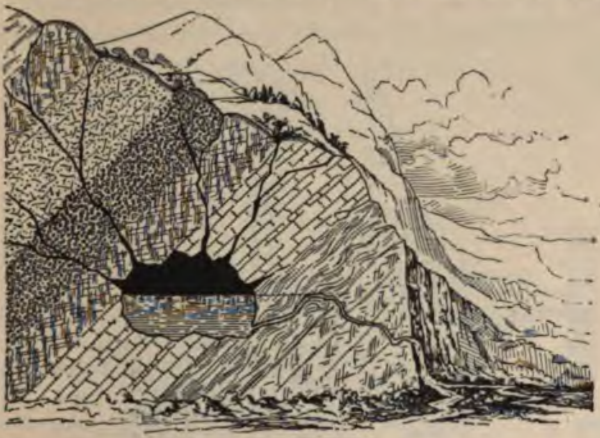

FIG. 202.

THE INJECTOR.

1076. A section of an injector is shown in Fig. 203. There are many different kinds of these instruments, but the principle is the same in all. When they are used for lifting water from a point below the discharge orifice and forcing it into the boiler of a steam engine or locomotive, they depend for their lifting action upon the creation of a partial vacuum by the action of steam. In the injector shown in Fig. 203, *F* is the connection for the steam pipe from the boiler, *P* is the connection for the pipe from the water supply, *N* is the connection to which the discharge pipe leading to the boiler is attached, and the waste water and steam are discharged through the **overflow** nozzle *O*.

The method of operation is as follows: The valve *B* is first opened by turning the wheel *W*; the **primer** valve *R* is then opened by the handle *J*, thus permitting steam to flow through the passage *E* and a connection, not shown in the figure, to the nozzle *u*. From *u* the jet of steam rushes out through *O*. A passage connects the chamber surrounding *u* with the space above the valve *L*. The jet of steam from *u* out through *O* carries with it the air in the chamber to which *O* is connected, thus forming a partial vacuum in the space above *L*; the air in the passages *D*, *C*, *G*, *H*, *K*, *T*, and in the water pipe connected at *P* is thus drawn out through the valve *L*, and a partial vacuum is formed, which permits the pressure of the atmosphere to force water through *P* until it finally fills the passages and flows out through *L* and the overflow nozzle *O*. As soon as water appears at *O*, the valve *R* is closed and the main steam valve *A* is opened by the wheel *S*, thus admitting steam to the passages *C*, *H*, *K*. This steam draws water from *G* through the opening surrounding *H* and discharges it through *K* with such a high velocity that it rushes past

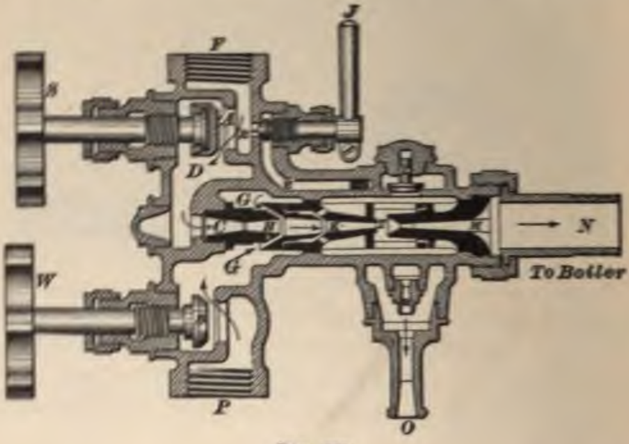

FIG. 203.

the opening *T* into the nozzle *M* and thence into the boiler.

THE LOCOMOTIVE BLAST.

1077. Fig. 204 shows the front end of a locomotive. *E* is the exhaust pipe, the center of which is directly in line with the center of the smokestack *S*. *T*, *T* are the tubes through which the hot furnace gases are discharged. The exhaust steam has a pressure of about two pounds above the atmosphere, and rushes through the exhaust pipe *E* and up the smokestack *S* with a very high velocity, taking the air out with it, and producing a partial vacuum in the space in front of the tubes. No air can get in this space except through the grates of the fire-box; consequently, this partial vacuum created in front of the tubes as described causes an influx of air through the grate, and produces the **forced draft,** or **blast.** The faster the engine runs, the greater the quantity of air drawn through the grate.

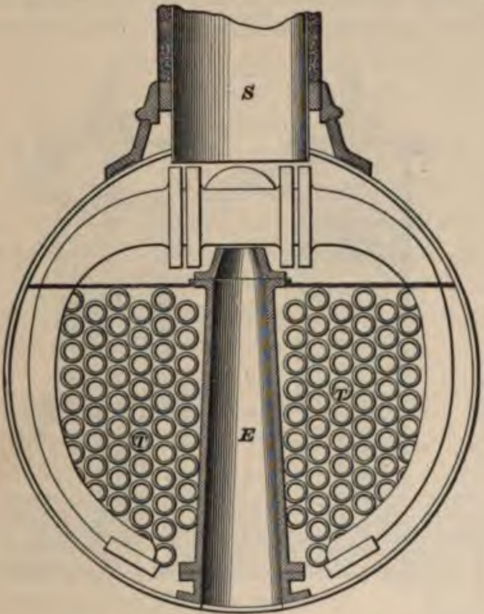

FIG. 204.

PUMPS.

1078. The Suction Pump.—A section of an ordinary suction pump is shown in Fig. 205. Suppose the piston to be at the bottom of the cylinder and to be just on the point of moving upwards in the direction of the arrow. As the piston rises it leaves a vacuum behind it, and the atmospheric pressure upon the surface of the water in the well causes it to rise in the pipe P, for the same reason that the mercury rises in the barometer tube. The water rushes up the pipe and lifts the valve V, filling the empty space in the cylinder B displaced by the piston. When the piston has reached the end of its stroke, the water entirely fills the space between the bottom of the piston and the bottom of the cylinder and also the pipe P. The instant that the piston begins its down stroke, the water in the chamber B tends to fall back into the well, and its weight forces the valve V to its seat, thus preventing any downward flow of the water. The piston now tends to compress the water in the chamber B, but this is prevented through the opening of the valves u, u

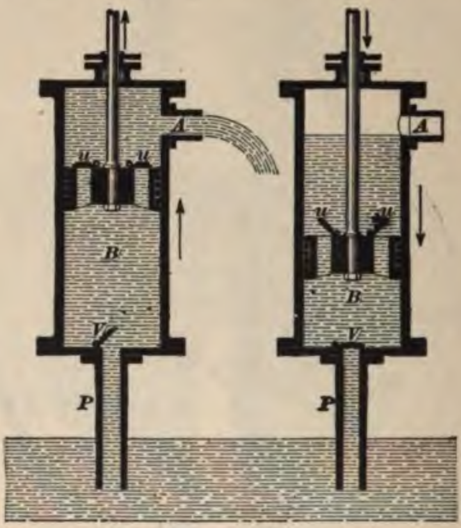

FIG. 205.

in the piston. When the piston has reached the end of its downward stroke, the weight of the water above closes the valves *u*, *u*. All the water resting on the top of the piston is then lifted with the piston on its upward stroke, and discharged through the spout *A*, the valve *V* again opening, and the water filling the space below the piston as before.

It is evident that the distance between the valve *V* and the surface of the water in the well must not exceed 34 feet, the highest column of water which the pressure of the atmosphere will sustain, since otherwise the water in the pipe would not reach to the height of the valve *V*. In practice this distance should not exceed 28 feet. This is due to the fact that there is a little air left between the bottom of the piston and the bottom of the cylinder, a little air leaks through the valves which are not perfectly air-tight, and a pressure is needed to raise the valve against its weight, which, of course, acts downwards. There are many varieties of the suction pump, differing principally in the valves and piston, but the principle is the same in all.

1079. The Lifting Pump.—A section of a **lifting pump** is shown in Fig. 206. These pumps are used when water is to be raised to greater heights than can be done with the ordinary suction pump. As will be perceived, it is essentially the same as the pump previously described, except that the spout is fitted with a cock and has a pipe attached to it, leading to the point of discharge. If it is desired to discharge the water at the spout, the cock may be opened; otherwise, the cock is closed, and the water is lifted by the piston up through the pipe *P'* to the point of discharge, the valve *c* preventing it from falling back into the pump, and the valve *V*

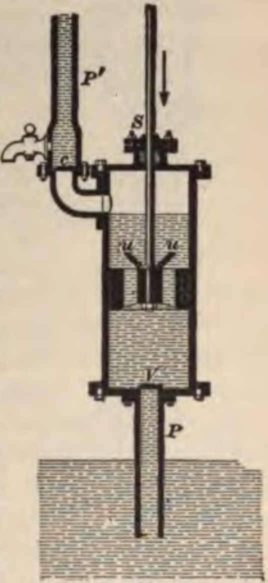

FIG. 206.

preventing the water in the pump from falling back into the well. It is not necessary that there should be a second pipe P', as shown in the figure, for the pipe P may be continued straight upwards, as shown in Fig. 207.

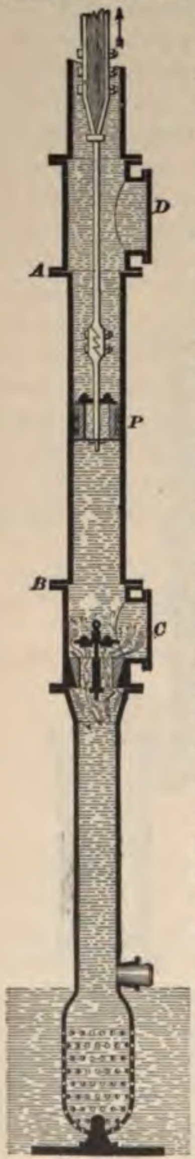

FIG. 207.

1080. In the figure is shown a section of a lifting pump for raising water from great depths, as from the bottom of mines to the surface. This pump consists of a series of pipes connected together, of which the lower end only is shown in the cut. That part of the pipe included between the letters A and B forms the pump cylinder in which the piston P works. That part of the pipe above the highest point of the piston travel, through which the water is discharged, is called the **delivery pipe,** and the part below the lowest point of the piston travel is called the **suction pipe.** The lower end of the suction pipe is expanded, and has a number of small holes in it, to keep out the solid matter. C is a plate covering an opening, and which may be removed to allow the suction valve to be repaired. D is a plate covering a similar opening through which the piston and piston valves may be repaired. The piston rod, or rather the piston stem, is made of wrought iron, inserted with wood, and connected with the piston. The only limit to the height to which a pump of this kind can raise water is the strength of the piston rod. Lifting pumps of this kind are used to raise water from great depths to the earth's surface; hence, a very long piston rod is necessary. In the lifting pump shown in Fig. 206 the water is raised from a point a few feet below the earth's surface to a point considerably higher. This requires the piston rod to move through a stuffing-box, as shown at *S*, and also necessitates the rod being round, in order that the water may not leak out.

1081. Force Pumps.—The **force pump** differs from the lifting pump in several important particulars, but chiefly in the fact that the piston is solid; that is, it has no valves. A section of a *suction and force pump* is shown in Fig. 208. The water is drawn up the suction pipe as before, when the piston rises; but when the piston reverses, the pressure on the water caused by the descent of the piston opens the valve *V'* and *forces* the water up the delivery pipe *P'*. When the piston again begins its upward movement, the valve *V'* is closed by the pressure of the water above it, and the valve *V* is opened by the pressure of the atmosphere on the water below it, as in the previous cases. For an arrangement of this kind, it is not necessary to have a stuffing-box. The water may be forced to almost any desired height. The force pump differs again from the lifting pump in respect to its piston rod, which should not be longer than is absolutely necessary in order to prevent it from *buckling*,

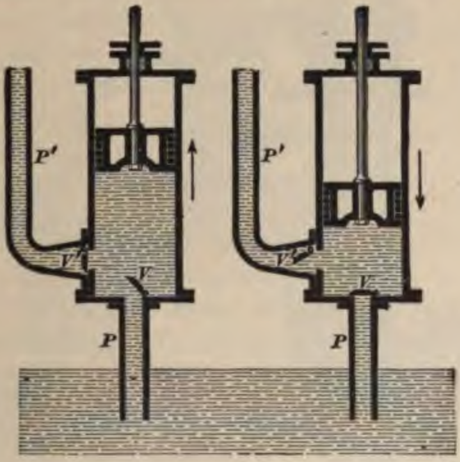

FIG. 208.

while in the lifting pump the length of the piston rod is a matter of indifference.

1082. Plunger Pumps.—When force pumps are used to convey water to great heights, the pressure of the water in the cylinder becomes so great that it becomes extremely difficult to keep the water from leaking past the piston, and the constant repairing of the piston packing becomes a nuisance. To obviate this difficulty the piston is made very long, as shown in Fig. 209, and is then called a **plunger.** The suction valve in this case consists of two clack valves inclined to each other and resting upon a square pin *A*; they are prevented from flying back too far during the up stroke of the plunger by the two uprights *I*, *I*. During the down stroke of the plunger the valves at *A* are closed and the delivery valve at *B* is open. A little air is always carried into the cylinder of a pump with the entering of the water. In force pumps this fact becomes a serious consideration, since, after repeated strokes, the air accumulates, and during the down stroke of the plunger it is compressed. After a time it would become sufficiently compressed to entirely prevent the water from entering through the suction valve, the pressure on the top of the valve being greater than that of the atmosphere below. In the pump shown in the figure, the plunger is a trifle smaller than the cylinder, and the air collects around the plunger below the stuffing-box. To remove this air a narrow passage *C*, shown by the dotted lines, that can be closed at its upper end by the cock *D*, connects the interior of the pump with the atmosphere when the cock is open. It is evident that this

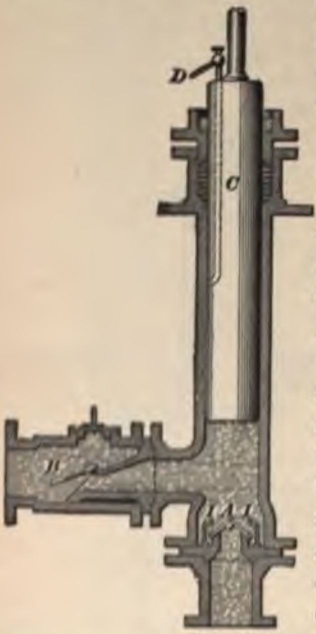

FIG. 209.

cock must not be opened, except during the down stroke of the plunger; for, if it were open during the up stroke, the pressure below the plunger being less than the pressure of the atmosphere above, the air would rush in instead of being expelled.

1083. Double-Acting Pumps.—In the pumps previously described, the discharge was intermittent; that is, the pump could only discharge when the piston was moving in one direction. In some cases it is necessary that there should be a continuous discharge; in all cases it takes more power to run the pump with an intermittent discharge, as a little consideration will show. If the height that the water is to be raised is considerable, its weight will be very great, and the entire mass must be put in motion during one stroke of the piston.

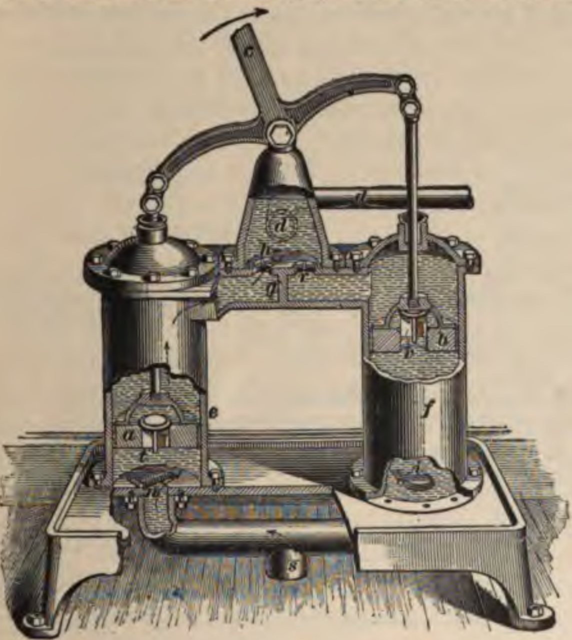

FIG. 210.

In order to obtain the advantage of a more continuous discharge, double-acting pumps are used. Fig. 210 shows a part sectional view of such a pump. Two pistons *a* and *b* are used, which are operated by one handle *c* in the manner shown. The pump has one suction pipe *s* and one discharge pipe *d*. The cylinders *e* and *f* are separated by a diaphragm *g*, so that they cannot communicate with each other above the pistons. In the figure, the handle *c* is moving to the right, the piston *a* upwards, and the piston *b* downwards. As the piston *a* moves upwards, it lifts the water above it and causes it to flow through the delivery valve *h* into the discharge pipe *d*. This upward movement of the piston creates a partial vacuum below it in the cylinder *e*, and causes the water to rush up the suction pipe *s* into the cylinder, as shown by the arrows. In the cylinder *f*, the downward movement of the piston *b* raises the piston valve *v*, and the weight of the water on the suction valve *i* keeps it closed. When the handle *c* has completed its movement to the right and begins its return, all of the valves on the right-hand side open except *v*, and those on the left-hand side close except *n*; water is then discharged into the delivery pipe by the cylinder *f*, and only at the instant of reversal is the flow into the delivery pipe *d* stopped.

1084. Air Chambers.—In order to obtain a continuous flow of water in the delivery pipe, with as nearly a uniform velocity as possible, an **air chamber** is usually placed on the delivery pipe of force pumps as near to the pump cylinder as the construction of the machine will allow. The air chambers are usually pear-shaped, with the small end connected to the pipe. They are filled with air which the water compresses during the discharge. During the suction, the air thus compressed expands and acts as an accelerating force upon the moving column of water, a force which diminishes with the expansion of the air, and helps to keep the velocity of the moving column more nearly uniform. An air chamber is sometimes placed upon the suction pipe. These air chambers not only tend to promote a uniform discharge, but they also equalize the stresses upon the pump, and prevent shocks due to the incompressibility of water.

They serve the same purpose in pumps that a fly-wheel does to the steam engine. Unless the pump moves very slowly, it is absolutely necessary to have an air chamber on the delivery pipe.

1085. Steam Pumps.—Steam pumps are force pumps operated by steam acting upon the piston of a steam engine, directly connected to the pump, and in many cases cast with the pump. A section of a double-acting steam pump showing the steam and water cylinders, with other details, is illustrated in Fig. 211. Here *G* is a steam piston, and *R* the piston rod, which is secured at its other end to the plunger *P*. *F* is a partition cast with the cylinder, which prevents the water in the left-hand half from communicating with that in the right-hand half of the cylinder. Suppose the piston to be moving in the direction of the arrow. The volume of the left-hand half of the pump cylinder will be increased by an amount equal to the area of the circumference of the plunger, multiplied by the length of the stroke, and the volume of the right-hand half of the cylinder will be diminished by a like amount. In consequence

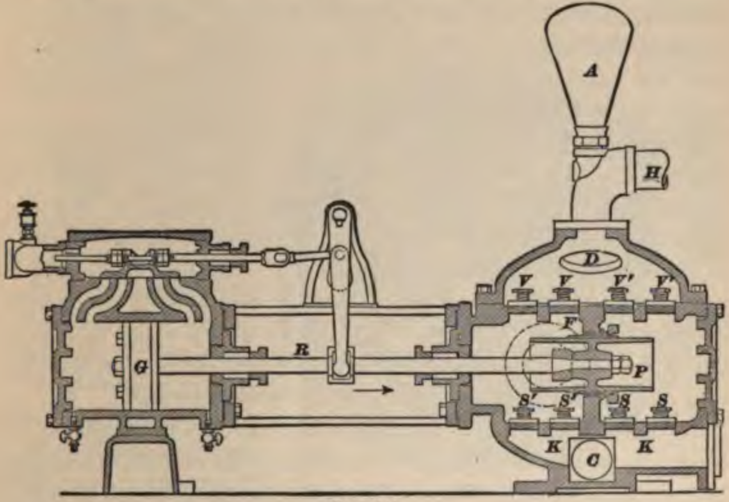

FIG. 211.

of this, a volume of water in the right-hand half of the cylinder equal to the volume displaced by the plunger in its forward movement will be forced through the valves *V'*, *V'* into the air chamber *A*, through the orifice *D*, and then discharged through the delivery pipe *H*. By reason of the partial vacuum in the left-hand half of the pump cylinder, owing to this movement of the plunger, the water will be drawn from the reservoir through the suction pipe *C* into the chamber *K*, *K*, lifting the valves *S'*, *S'*, and filling the space displaced by the plunger. During the return stroke the water will be drawn through the valves *S*, *S* into the right-hand half of the pump cylinder, and discharged through the valves *V*, *V* in the left-hand half. Each one of the four suction and four discharge valves is kept to its seat, when not working, by light springs, as shown.

There are many varieties and makes of steam pumps, the majority of which are double-acting. In many cases two steam pumps are placed side by side, having a common delivery pipe. This arrangement is called a **duplex pump.** It is usual to so set the steam pistons of duplex pumps that when one is completing the stroke the other is in the middle of its stroke. A double-acting duplex pump made to run in this manner, and having an air chamber of sufficient size, will deliver water with nearly a uniform velocity.

In mine pumps for forcing water to great heights, the plungers are made solid, and in most cases extended through the pump cylinder. In many steam pumps pistons are used instead of plungers, but when very heavy duty is required plungers are preferred.

1086. Centrifugal Pumps.—Next to the direct-acting steam pump, the **centrifugal pump** is the most valuable instrument for raising water to great heights that has yet been described. As the name denotes, the effects produced by centrifugal force are made use of. Fig. 212 represents one with half of the casing removed. The hub *S* is hollow, and is connected directly to the suction pipe. The curved arms *a*, called **vanes** or **wings,** are revolved with a high velocity in the direction of the arrow, and the air enclosed between them is driven out through the discharge passage and delivery pipe *D D*. This creates a partial vacuum in the casing and suction pipe, and causes the water to flow in through *S*. This water is also made to revolve with the vanes, and, of course, with the same velocity. The centrifugal force of the revolving water causes it to fly outwards towards the end of the vanes, and becomes greater the farther away it gets from the center. This causes it to leave the vanes, and finally to leave the pump by means of the discharge passage and delivery pipe *D D*. The height to which the water can be forced depends upon the velocity of the revolving vanes. In the construction of the centrifugal pump, particular care is required in giving the correct form to the vanes; the efficiency of the machine depends greatly upon this point being attended to. What is required is to raise the water, and the energy used to drive the pump should be devoted as far as possible to this one purpose. The water when it is raised should be delivered with as little velocity as possible, for any velocity which the water then possesses, has been produced at the expense of the energy used to drive the pump. The form of the vanes is such that the water is delivered at the desired height with the least expenditure of energy.

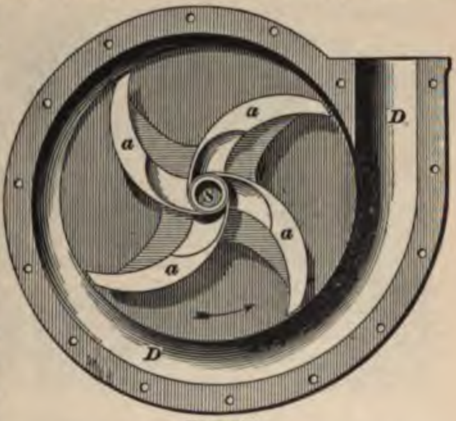

FIG. 212.

The number of vanes depends upon the size and capacity of the pump. It will be noticed that, in the pump shown in the figure, the vanes have sharp edges near the hub. The object of this is to provide for a free ingress of the water, and also to cut any foreign substance that may enter the pump and prevent it from working properly.

Almost any liquid can be raised with these pumps, but when used for pumping chemicals, the casing and vanes are made of materials that the chemicals will not affect.

1087. The Hydraulic Ram.—The construction of a hydraulic ram is shown in Fig. 213. This machine is used for raising water from a point below the level of the water in a spring or reservoir to a point considerably higher, with no power other than that afforded by the inertia of a moving column of water. In the figure, *a* is a pipe called the *drive pipe*, connecting the ram with the reservoir; the valve *b* slides freely in a guide, and is provided with lock-nuts to regulate the distance that the valve can fall below its seat. When the water is first turned on by opening the valve *n*, the valve *b* is already opened, and the water flows out through *c*, as shown. As the discharge continues, the velocity of the water in the drive pipe will increase until the upward pressure against the valve *b* is sufficient to force the valve to its seat. The actual closing of the valve takes place very suddenly, and the momentum of the column of water, which was moving with an increasing velocity through the drive pipe *a*, will very rapidly force some water through the valve *d* into the air-chamber *f*. Immediately after this, a rebound takes place, and for a short interval of time the water flows

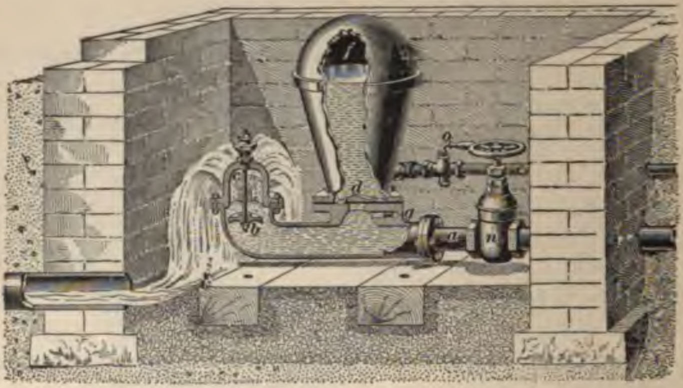

FIG. 213.

back up the drive pipe *a* and tends to form a vacuum under the air chamber valve *d*; this opens the snifter valve *g* and admits a little air, which accumulates under the valve *d* and is forced into the air chamber with the next shock. This air keeps the air chamber constantly charged; otherwise, the water, being under a greater pressure in the air chamber than it is in the reservoir, would soon absorb the air in the chamber and the ram would cease to work until the chamber was recharged with air. The rebound also takes the pressure off the under side of valve *b* and causes it to drop, and the above-described operations are repeated. The delivery pipe is shown at *e*; a steady flow of water is maintained through it by the pressure of the air in the chamber *f*; this air also acts as a cushion when valve *b* suddenly closes, and prevents undue shock to the parts of the ram.

The height to which water can be raised by the hydraulic ram depends upon the weight of the valve *b* and the velocity of the water in *a*.

1088. Power Necessary to Work a Pump:

Rule I.—*In all pumps, whether lifting, force, steam, single- or double-acting, or centrifugal, the number of foot-pounds of power needed to work the pump is equal to the weight of the water in pounds, multiplied by the vertical distance in feet between the level of the water in the well, or source, and the point of discharge, plus the work necessary to overcome the friction and other resistances.*

Rule II.—*The work done in one stroke of a pump is equal to the weight of a volume of water equal to the volume displaced by the piston during the stroke, multiplied by the total vertical distance in feet through which the water is to be raised, plus the work necessary to overcome the resistances.*

A little consideration will make Rule II evident. Suppose that the height of the suction is 25 feet; that the vertical distance between the suction valve and the point of discharge is 100 feet; that the stroke of the piston is 15 inches, and that its diameter is 10 inches. Let the diameters of the suction pipe and delivery pipe be 4 inches each. The volume displaced by the pump piston or plunger in one stroke equals $\frac{10^2 \times .7854 \times 15}{1728} = .68177$ cubic foot. The weight of an equal volume of water $= .68177 \times 62.5 = 42.61063$ pounds. Now, in order to discharge this water, *all* of the water in the suction and delivery pipes had to be moved through a certain distance in feet equal to .68177 divided by the area of the pipes in square feet.

Four inches $= \frac{1}{3}$ of a foot. $(\frac{1}{3})^2 \times .7854 = \frac{.7854}{9} = .0872\frac{2}{3}$ square foot. $.68177 \div .0872\frac{2}{3} = 7.8125$ feet.

The weight of the water in the delivery pipe is $(\frac{1}{3})^2 \times .7854 \times 100 \times 62.5 = 545.42$ pounds.

The weight of the water in the suction pipe is $(\frac{1}{3})^2 \times .7854 \times 25 \times 62.5 = 136.35$ pounds.

$545.42 + 136.35 = 681.77$ pounds = the total weight of water moved in one stroke. The distance that it is moved in one stroke is 7.8125 feet. Hence, the number of foot-pounds necessary for one stroke is $681.77 \times 7.8125 = 5{,}326.33$ foot-pounds. Had this result been obtained by Rule II, the process would have been as follows: The weight of the water displaced by the piston in one stroke was found to be 42.61063 pounds. $42.61 \times 125 = 5{,}326.33$ pounds, which is exactly the same as the result obtained by the previous method, and is a great deal shorter.

EXAMPLE.—What must be the necessary horsepower of a double-acting steam pump if the vertical distance between the point of discharge and the point of suction is 96 feet? The diameter of the pump cylinder is 8 inches, the stroke is 10 inches, and the number of strokes per minute is 120. Allow 25% for friction, etc.

SOLUTION.—Since the pump is double-acting, it raises a quantity of water equal to the volume displaced by the plunger at every stroke. The weight of the volume of water displaced in one stroke $= (\frac{8}{12})^2 \times .7854 \times \frac{10}{12} \times 62.5 = 18.18$ pounds, nearly.

$18.18 \times 96 \times 120 = 209{,}433.6$ foot-pounds per minute.

Since 25% is to be allowed for friction, the actual number of foot-pounds per minute $= 209{,}433.6 \div .75 = 279{,}244.8$ foot-pounds per minute.

One horsepower = 33,000 foot-pounds per minute; hence, $\frac{279{,}244.8}{33{,}000} = 8.462$ H. P., nearly. Ans.

HEAT.

THE PROPERTIES, SOURCES, AND MEASUREMENT OF HEAT.

1089. The Nature of Heat.—As to the exact nature of heat, scientists differ, but all modern thinkers and investigators agree that *heat is a form of energy*, and that it *is a kind of motion.* It is not purposed here to enter into the different theories regarding heat, but as much of the generally accepted theory will be given as will be necessary to make clear the principles which are to follow.

In Art. **831** it was stated that bodies were composed of molecules. Notwithstanding the extreme minuteness of the molecules, they play a very important part in the modern theory of heat. Each molecule attracts the molecules surrounding it in a manner similar to the attraction between the earth and bodies near its surface, only with an immensely greater force in proportion to their sizes. Without going into any theory regarding the precise nature of heat, it will be taken for granted that each and every molecule has a rapid vibratory motion to and fro, and that the molecules are kept from getting beyond a certain distance from one another by the attractive force between them. This attractive force is called **cohesion;** without it, everything throughout the universe would crumble instantly into the finest dust.

In Art. **846** it was stated that the molecules were supposed to be round; it is likewise supposed that they are at a considerable distance apart, compared with the diameter of the molecule. When heat is applied to a body the number of these vibrations is greatly increased, proportional to the amount of heat supplied. In consequence of this increase, the distance through which a molecule moves is increased, and the force of cohesion which binds them together is lessened. If enough heat is added to a solid, the force of cohesion is so far overcome that the body melts. If more heat is supplied in sufficient quantity, the melted body becomes a vapor, and so long as it is kept at this temperature the force of cohesion has no effect, in consequence of the number of vibrations having been so far increased and the distance between any two molecules having become too great for the force of cohesion to act. If the vapor be cooled, the number of vibrations will decrease, and also the distance between any two molecules; the force of cohesion begins to act, and the body becomes a liquid. If cooled further, and a sufficient quantity of heat is removed—in other words, if the number of vibrations is so far decreased that the molecules are comparatively near together—the body becomes a solid and remains so until the temperature is again increased to the melting point.

1090. If a body is heated and brought near the hand, the sensation of warmth is felt; if heat be removed from this same body, and it is again brought near the hand, the sensation of cold is felt. The heat which thus manifests itself is called **sensible heat,** because any change from any state to a hotter or colder state is indicated at once by the sense of feeling, or by the aid of instruments called **thermometers.** The more sensible heat a body possesses, the hotter it is; the more sensible heat that is taken away from it, the colder it is.

THERMOMETERS.

1091. The different states that a body is in according to the amount of sensible heat it possesses are indicated by the word **temperature,** and by comparison with some other body having the same amount of sensible heat. Thus, a piece of iron having exactly the same amount of sensible heat as a piece of melting ice, is said to have *the temperature of melting ice.* If a piece of lead has the same amount of sensible heat as a kettle of boiling water, it is said to have *the temperature of boiling water*, etc.

1092. Owing to the imperfection of our senses, it is impossible to determine by their aid, with any degree of accuracy, the temperature of different bodies; hence, for this purpose, thermometers are used. In these instruments the effects of heat upon bodies are made use of in obtaining the temperature, the most common method being to utilize the expansive effect of heat upon liquids. Liquids are used for ordinary purposes in place of solids or gases, because in the first the expansion is too small, and in the second too great. Mercury and alcohol are the only liquids used—the former because it boils only at a very high temperature, and the latter because it does not solidify at the greatest known cold produced by ordinary means.

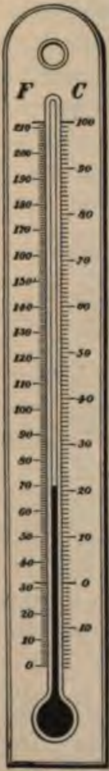

FIG. 214.

1093. In Fig. 214 is shown a mercurial thermometer with two sets of graduations on it. The one on the left, marked *F*, is the **Fahrenheit thermometer,** so named after its inventor, and is the one commonly used in this country and in England; the one on the right, marked *C*, is the **Centigrade thermometer,** and is used by scientists throughout the world, on account of the graduations being better adapted for calculations. As will be seen, the instrument consists of a glass tube having a bulb at one end and closed at the other, so as to keep the air out. Before closing the upper end, the tube is partially filled with mercury, and the air above it is driven out by heating the mercury to near its boiling point, when the tube above the mercury will be filled with mercurial vapor. It is now sealed, and, on cooling, the vapor condenses and a vacuum results. The expansion or contraction of the mercury, by applying or withdrawing heat from the body with which the bulb is in contact, causes the highest point of the mercury column to rise or fall, and, since for equal changes of temperature the mercury rises or falls equal distances, this instrument, when properly made and graduated, indicates any change in temperature with great accuracy.

1094. For a good thermometer, the inside diameter should be the same throughout its length. In order to graduate the thermometer, it is placed in melting ice, and the point to which the mercurial column falls is marked **freezing.** It is then placed in the steam rising from water boiling in an open vessel, and the point to which the mercurial column rises is marked **boiling.**

1095. There are now two fixed points, the freezing point and the boiling point. If it is desired to make a Fahrenheit thermometer, the distance between these two fixed points is divided into 180 parts, called degrees. The freezing point is marked 32°, and the boiling point 212°. Thirty-two parts are marked off from the freezing point downwards, and the last one is marked 0°, or zero. The graduations are carried above the boiling point and below the zero point as far as desired. This thermometer was invented in 1714, and was the first to come into general use.

1096. In graduating a Centigrade thermometer, the freezing point is marked 0°, or zero, and the boiling point 100°; the distance between the freezing and boiling points is divided into 100 equal parts; these equal divisions are carried as far below the freezing point and above the boiling point as desired. The reason that Fahrenheit placed the zero point on his thermometer 32° below freezing was because that was the lowest temperature he could obtain, and he supposed that it was impossible to obtain a lower one. Where there is any doubt as to the thermometer used, the first letter of the name is placed after the degree of temperature. For example, 183° F. means 183° above zero on the Fahrenheit instrument; 183° C. would mean 183° above zero on the Centigrade instrument.

1097. In Russia and a few other countries another instrument is used, called the **Reaumur;** the freezing point is marked 0°, or zero, and the boiling point. 80°, the space between these two points being divided into 80 equal parts; 183° R. would mean 183° on the Reaumur thermometer.

1098. Of these three thermometers, the Centigrade is used the most; but, since the Fahrenheit instrument is the one in general use in this country, all temperatures given here will be understood to be in Fahrenheit degrees, unless otherwise stated. In order to distinguish the temperatures below the zero point from those above, the sign of subtraction is placed before the figures, indicating the number of degrees below zero. Thus, $-18°$ C. would mean that the temperature was 18° below the zero point on the Centigrade thermometer; $-25.4°$ F. would mean 25.4° below zero on the Fahrenheit thermometer. As was stated in Art. **1056,** the point of absolute zero, or $-460°$ F., is the point at which all vibratory motion of the molecules ceases. It is supposed that, at this temperature, and under a heavy pressure so as to bring the molecules close enough together, even hydrogen would be solidified. The absolute zero on the Centigrade scale is $-273\frac{1}{3}°$ C.

1099. The **absolute temperature** is the temperature measured above the point of absolute zero. Hence, on the Fahrenheit scale, the absolute temperature T is $460° + t°$ when $t =$ the ordinary temperature, and is above zero. If $t°$ is below zero, its value is negative, and the absolute temperature T is $460° + (-t°) = 460° - t°$.

Throughout the remainder of this volume, where temperatures are mentioned, t will denote the ordinary temperature indicated by the thermometer, and T the absolute temperature.

EXAMPLE.—What are the absolute temperatures of 212°, 32°, and $-39.2°$?

SOLUTION.—Since no scale is specified, the Fahrenheit is the one intended to be used.

$$460° + 212° = T = 672°. \quad \text{Ans.}$$
$$460° + 32° = T = 492°. \quad \text{Ans.}$$
$$460° - 39.2° = T = 420.8°. \quad \text{Ans.}$$

1100. The absolute temperature on the Centigrade scale is $T = 273\frac{1}{3}^\circ + t^\circ$ when t° is above zero, or $T = 273\frac{1}{3}^\circ - t^\circ$ when t° is below zero.

Example.—What are the absolute temperatures corresponding to 100°, 4°, and -40° C.?

Solution.—$273\frac{1}{3}^\circ + 100^\circ = T = 373\frac{1}{3}^\circ$ C.
$273\frac{1}{3}^\circ + 4^\circ = T = 277\frac{1}{3}^\circ$ C.
$273\frac{1}{3}^\circ - 40^\circ = T = 233\frac{1}{3}^\circ$ C.

1101. It is frequently necessary to change from one scale to the other. For example, what would 80° C. be on the Fahrenheit scale?

Since the number of degrees between the freezing point and boiling point on the Centigrade scale is 100, and on the Fahrenheit 180, it is evident that if F = the number of degrees Fahrenheit, and C = the number of degrees Centigrade, that

$$F : C :: 180 : 100, \text{ or } F = \tfrac{180}{100}\,C = \tfrac{9}{5}\,C.$$

Also, $C = \frac{100}{180}\,F = \frac{5}{9}\,F$. Therefore,

1102. To change Centigrade temperatures into their corresponding Fahrenheit values:

Rule.—*Multiply the temperature Centigrade by* $\frac{9}{5}$, *and add 32°; the result will be the temperature Fahrenheit.*

1103. To change Fahrenheit temperatures into their corresponding Centigrade values:

Rule.—*Subtract 32° from the temperature Fahrenheit, and multiply by* $\frac{5}{9}$, *and the result will be the temperature Centigrade.*

1104. Expressing these two rules by means of formulas, let t_c = temperature Centigrade, and t_f = temperature Fahrenheit. Then,

$$t_f = \tfrac{9}{5}\,t_c + 32^\circ, \qquad \textbf{(65.)}$$

$$\text{and } t_c = (t_f - 32^\circ)\,\tfrac{5}{9}. \qquad \textbf{(66.)}$$

Example.—Change (*a*) 100° *C.*, (*b*) 4° *C.*, and (*c*) —40° *C.* into Fahrenheit temperatures.

Solution.—(*a*) $t_f = \frac{9}{5}\,t_c + 32 = \frac{9}{5} \times 100 + 32 = 212^\circ$ *F.* Ans.
(*b*) $t_f = \frac{9}{5} \times 4 + 32 = 39.2^\circ$ *F.* Ans.
(*c*) $t_f = \frac{9}{5} \times -40 + 32 = -40^\circ$ *F.* Ans.

Example.—Change (*a*) 60° *F.*, (*b*) 32° *F.*, and (*c*) —20° *F.* into their corresponding Centigrade temperatures.

Solution.—(*a*) $t_c = (t_f - 32)\frac{5}{9} = (60 - 32)\frac{5}{9} = 15\frac{5}{9}°$ *C.* Ans.

(*b*) $t_c = (32 - 32)\frac{5}{9} = 0°$ *C.* Ans.

(*c*) $t_c = (-20 - 32)\frac{5}{9} = -28\frac{8}{9}°$ *C.* Ans.

1105. Since mercury freezes at —37.84° F. (this corresponds to —38.8° C.), some other means must be had to obtain temperatures below this point. For this purpose alcohol is used in place of mercury. This liquid has never been frozen until very recently, and then only at an extremely low temperature. Since alcohol vaporizes at 173° F., the boiling point of water cannot be marked on the alcohol thermometer by heating it to that point. The freezing point is determined as for mercury. An alcohol and a mercurial thermometer are placed in a vessel containing hot water or other liquid, and the point to which the alcohol column rises is marked. Suppose that the point to which the mercury column rises is marked 132°, then the distance between the point marked and the freezing point would be divided into 132 — 32 = 100 equal parts, and each one of these parts would correspond to one degree on the mercurial thermometer. These equal divisions are then carried below the zero point as far as it is desired.

1106. There are many other kinds of thermometers, some of which depend upon the expansion and contraction of different metals and gases when heated and cooled. For temperatures above 662° F., the point at which mercury vaporizes, other means are employed to obtain the temperatures.

EXPANSION OF BODIES.

1107. The volume of any body, solid, liquid, or gaseous, is always changed if the temperature is changed; nearly all of them expand when heated, and contract when cooled. In solids, which have definite figures, the expansion may be considered in three ways, according to the conditions: 1st.—The expansion in one direction, as the elongation of an iron bar; this is called **linear expansion**. 2d.—**Surface expansion,** where the area is increased. 3d.—**Cubical expansion,** where the increase in the whole volume is considered.

1108. In Fig. 215 is shown an apparatus for exhibiting the linear expansion of a solid body. A metal rod *A* is fixed at one end by a screw *B*, the other end passing freely through the eye *C*, held in the post, and pressing against the short arm of the indicator *F*. The rod is heated as shown, and its elongation causes the indicator to move along the arc *D E*.

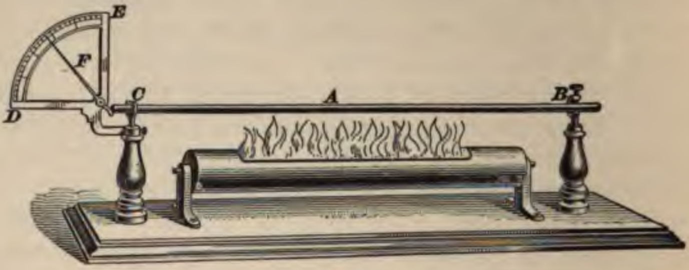

FIG. 215.

1109. An illustration of surface expansion is afforded nearly every day in machine shops, particularly in locomotive shops, where piston rods, crank-pins, etc., are "shrunk in" and tires shrunk on their centers. In shrinking on a tire, it is bored a little smaller than the wheel center. The tire is then heated until the area of its circumference is expanded enough to allow it to slide over the wheel center. It is then cooled with cold water, when it contracts, tending to regain its original area, but is prevented by reason of the wheel center being a trifle larger. This causes the tire to hug the center with immense force and prevents it from coming off.

1110. Cubic expansion may be illustrated by means of a *Gravesandes' ring*. This consists of a brass ball *a*, Fig.

216, which at ordinary temperatures passes freely through the ring *m*, of very nearly the same diameter. When the ball is heated, it expands so much that it will no longer pass through the ring.

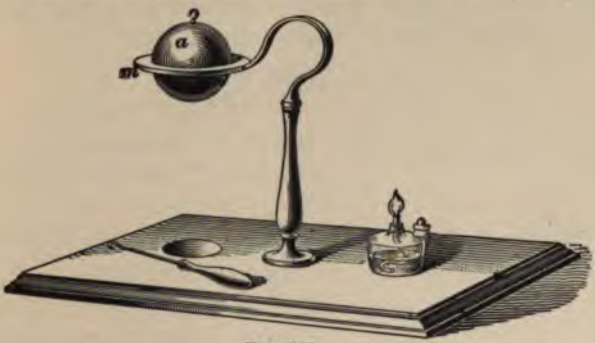

FIG. 216.

1111. The expansion of liquids is clearly shown in the mercurial and alcohol thermometers. The expansion of gases was treated on to some extent in pneumatics.

COEFFICIENT OF EXPANSION.

1112. Suppose that the temperature of the metal rod, shown in Fig. 215, was 32° F. before heating, and exactly 10 feet long; that after the temperature had been raised 1°, or to 33°, the bar was 10 ft. + $\frac{1}{1200}$ in. long. The linear expansion would then be (10 ft. + $\frac{1}{1200}$ in.) − 10 ft. = $\frac{1}{1200}$ in., and the ratio between this expansion and the original length of the bar would be

$$\tfrac{1}{1200} : 10 \times 12, \text{ or } \tfrac{1}{1200 \times 120} : 1, \text{ or } .000006944 : 1.$$

For every increase of temperature of 1° this rod would elongate .000006944 of its length. This number .000006944, which equals the expansion of the rod for one degree rise of temperature divided by the original length, is called the **coefficient of linear expansion.** Had the temperature of the rod been increased 100° instead of 1°, the amount of elongation would have been $.000006944 \times 100 = .0006944$, of its length, or $.0006944 \times 120 = .083328''$, or $\frac{1}{12}''$. Table 19 contains the coefficients of expansion for a number of solids, mercury, and alcohol, and the average cubical expansion of gases. No liquids are given except mercury and alcohol, for the reason that the coefficient of expansion for liquids is different for different temperatures.

TABLE 19.

Name of Substance.	Linear Expansion.	Surface Expansion.	Cubic Expansion.
Cast Iron	.00000617	.00001234	.00001850
Copper	.00000955	.00001910	.00002864
Brass	.00001037	.00002074	.00003112
Silver	.00000690	.00001390	.00002070
Bar Iron	.00000686	.00001372	.00002058
Steel (untempered)	.00000599	.00001198	.00001798
Steel (tempered)	.00000702	.00001404	.00002106
Zinc	.00001634	.00003268	.00004903
Tin	.00001410	.00002820	.00003229
Mercury	.00003334	.00006668	.00010010
Alcohol	.00019259	.00038518	.00057778
Gases	....	....	.00203252

1113. Let L = length of any body;

l = amount of expansion or contraction due to heating or cooling the body;

A = area of any section of the body;

a = increase or decrease of area of the same section after heating or cooling the body;

V = volume of the body;

v = increase or decrease in volume due to heating or cooling the body;

C_1 = coefficient of expansion taken from column 1, Table 19;

C_2 = coefficient taken from column 2, Table 19;

C_3 = coefficient taken from column 3, Table 19;

t = difference in degrees of temperature between the original temperature and the temperature of the body after it has been heated or cooled.

Then, $l = L\,C_1\,t.$ **(67.)**

$a = A\,C_2\,t.$ **(68.)**

$v = V\,C_3\,t.$ **(69.)**

EXAMPLE.—How much will a bar of untempered steel, 14 ft. long, expand if its temperature is raised 80° ?

SOLUTION.—Since only one dimension is given, that of length, linear expansion only can be considered. From Table 19 the coefficient of linear expansion per unit of length for a rise in temperature of 1° is found to be .00000599 for untempered steel. Hence, using formula **67**, $l = L\,C_1\,t$, and substituting $14 \times .00000599 \times 80 = .0067088$ ft., or $.0067088 \times 12 = .0805056$ in.

This seems to be a very small amount, but in engineering constructions, where long pieces are rigidly connected, it must be taken into account. If the cross-section of the above bar were 2 in. square, and the bar was fitted tightly between two supports, an expansion of the above amount would exert a pressure against the supports of about 58,000 pounds.

Suppose that an iron rod $1\frac{1}{2}$ inches in diameter and 100 feet long was used as a tie-rod in constructing a bridge; that it was put in place and securely fastened to two rigid supports during a warm day in summer when the temperature in the sunlight was, say, 110°. On a cold day in winter, when the thermometer registered zero, the amount that the bar would tend to shorten, owing to this change in temperature, would be, substituting these values in formula **67**,

$$.00000686 \times 100 \times 110 = .07546 \text{ ft.} = .90552 \text{ in.}$$

If this rod were rigidly secured, so that it could neither stretch nor shorten, it would then exert a pull on the supports of about 33,400 pounds.

EXAMPLE.—The wheel center of a locomotive driver is turned to exactly 50″ in diameter. If the steel tire be bored 49.94″ in diameter, to what temperature must the tire be raised in order that it may be easily shoved over the center ? Assume that the diameter of the tire is expanded to $\frac{1}{1000}$ of an inch larger than the center, and that the original temperature is 60°.

SOLUTION.—For this case formula **68** may be used. The original diameter of the tire is 49.94 in., and it is to be increased to 50.001″. The area of a circle 49.94″ in diameter is 1,958.79 sq. in.; area of a circle 50.001″ in diameter is 1,963.58 sq. in. The difference between them is $1,963.58 - 1,958.79 = 4.79$ sq. in. $= a$ in formula **68.** Hence, since $C_2 = .00001198$, and $A = 1,958.79$, substitute these values in $a = A\, C_2\, t$, and $4.79 = 1,958.79 \times .00001198 \times t = .023466\, t$. Therefore, $t = \frac{4.79}{.023466} = 204.125°$, and $204.125° + 60° = 264.125°$. Ans.

NOTE.—Owing to the form of the equation here denoted by formula **68,** and to the manner in which the coefficients C_2 were determined, this example may be more easily solved by means of formula **67.** Thus, regard the diameter as a linear dimension and apply formula **67.** Increase in diameter $= l = 50.001 - 49.94 = .061''$. $L = 49.94$ and $C_1 = .00000599$. Substituting $.061 = 49.94 \times .00000599 \times t$, or $t = \frac{.061}{49.94 \times .00000599} = 203.92°$, and $203.92° + 60° = 263.92°$. Ans.

The slight difference in the two results is immaterial, and was to have been expected.

EXAMPLE.—What is the decrease in volume of a copper cylinder 30″ long and 22″ in diameter if cooled from 212° to 0°, the measurement being taken at a temperature of 70°?

SOLUTION.—$212° - 70° = 142°$ = the increase in temperature above 70°. Use formula **69,** $v = V\, C_3\, t$.

$$V = 22^2 \times .7854 \times 30 = 11,404 \text{ cu. in.}$$
$$v = 11,404 \times .00002864 \times 142 = 46.38 \text{ cu. in.}$$
$$11,404 + 46.38 = 11,450.38 \text{ cu. in.} = \text{the volume at } 212°.$$
$$70° - 0° = 70° = \text{the difference in temperature.}$$
$$v = 11,404 \times .00002864 \times 70 = 22.86 \text{ cu. in., nearly.}$$
$$46.38 + 22.86 = 69.24 \text{ cu. in.} \quad \text{Ans.}$$

The bars of a furnace must not be fitted tightly at their extremities, but must be free at one end; otherwise, in expanding they would split the masonry.

In laying the rails on railways, a small space is left between the successive rails; for, if they touched, the force of expansion would cause them to curve or to break the chairs. Water-pipes are fitted to one another by means of telescope joints, which allow room for expansion; so, also, are steam pipes, by means of the so-called expansion joints. If a glass vessel is heated or cooled too rapidly, it cracks, especially if it is thick; the reason for this is that, since glass is a poor conductor of heat, the sides become unequally heated, and, consequently, unequally expanded, which causes a fracture.

1114. It will be found, upon trial, that the three preceding formulas will not work back; i. e., if the length of a bar, after it has been heated, be found by formula **67,** and an attempt be made to reduce the bar to its original length by again applying formula **67,** and substituting for t the same value as in the first case, the value obtained for l will be slightly different in the two cases. The difference, however, is so slight that it is neglected in practice. If, however, the student desires to obtain exactly the same result in both cases, he must use the following more cumbersome formula, in which t_1, t_2, l_1, l_2, are, respectively, the original and final temperatures, the original and final lengths, and C_1 has the same value as in formula **67:**

$$l_2 = \left[\frac{1 + C_1\ (t_2 - 32)}{1 + C_1\ (t_1 - 32)}\right] l_1. \qquad \textbf{(70.)}$$

This formula is always used when calculating the expansion of gases by substituting V_1, V_2, and $\frac{1}{492}$ for l_1, l_2, and C_1, respectively. The results obtained will be exactly the same as those obtained by formula **58,** Art. **1054.** For, substituting the values as directed, the formula becomes

$$V_2 = \left[\frac{1 + \frac{1}{492}\ (t_2 - 32)}{1 + \frac{1}{492}\ (t_1 - 32)}\right] V_1 = \frac{\frac{492 + t_2 - 32}{492}}{\frac{492 + t_1 - 32}{492}} \times V_1 = \left(\frac{460 + t_2}{460 + t_1}\right) V_1.$$

1115. Although, as stated before, solids and liquids expand very nearly uniformly throughout all ranges of temperature, water is a marked exception to the general rule. If water is cooled down from its boiling point, it continually contracts until it reaches 39.2° F., when it begins to expand, until it freezes at 32° F. On the other hand, if water at 32° F. is heated, it contracts until it reaches 39.2° F., when it commences to expand. Therefore, the density of water is greatest where this change occurs. The importance of this exception is seen in the fact that ice forms on the *surface* of water, since it is lighter than the warmer body of water lying at varying depths below it. Were it not for this fact, all the large bodies of water would freeze solid, and would so affect the climate of the earth that it would be uninhabitable. The coefficient of expansion of water is such a very changeable quantity (varying with the temperature) that a special table is necessary.

1116. The effect of heat upon the expansion of gases was treated of in Arts. **1056,** etc., and will not be repeated here. It should be stated, however, that the constant .37052, used in formulas **60** and **61,** Arts. **1056** and **1057,** has that value only for air. For other gases it varies. If the value of this constant for any gas be represented by R, formula **61,** Art. **1057,** becomes

$$p\,V = R\,W\,T. \qquad \textbf{(71.)}$$

The value of R for several gases is given in Table 20.

TABLE 20.

Gas.	Volume of One Pound at 32° F. and a Tension of 1 Atmosphere (14.7 lb. per sq. in.).	Weight of One Cu. Ft. at 32° F. and a Tension of 1 Atmosphere (14.7 lb. per sq. in.).	R.
Air	12.388	.08073 lb.	.37052
Oxygen	11.2056	.08925 lb.	.33552
Nitrogen	12.7226	.0786 lb.	.38143
Hydrogen	178.891	.00559 lb.	5.34946

EXAMPLE.—What is the volume of 3 ounces of hydrogen gas having a tension of 20 pounds per square inch and a temperature of 80°?

SOLUTION.—3 ounces = $\frac{3}{16}$ of a pound. Since $t = 80°$, $T = 460 + 80 = 540°$. $R = 5.34946$ from Table 20. Hence, by formula **71,** $p\,V = R\,W\,T$, or $20\,V = 5.34946 \times \frac{3}{16} \times 540 = 541.6328$, and $V = \frac{541.6328}{20} = 27.08164$ cu. ft.; say, 27.082 cu. ft. Ans.

Example.—What is the weight of 10 cu. ft. of oxygen having a tension of one atmosphere and a temperature of 60° ?

Solution.—By formula **71**, $p\ V = R\ W\ T$, or $10 \times 14.7 = .33552 \times W \times 520$. Hence, $147 = 174.4704\ W$,

$$\text{and } W = \frac{147}{174.4704} = .84255 \text{ lb.} \quad \text{Ans.}$$

In Table 19 the coefficient of expansion for gases was given as .00203252; this is the fraction $\frac{1}{492}$ reduced to a decimal. This value of the coefficient of expansion is very nearly the same for all gases, particularly so for those which are very difficult to liquefy.

EXAMPLES FOR PRACTICE.

1. What are the absolute temperatures corresponding to (*a*) 120° R., (*b*) 120° C., and (*c*) 120° F. ?

Ans. (*a*) 338⅔° R.; (*b*) 393⅓° C.; (*c*) 580° F.

2. Change — 10° R. to the corresponding Fahrenheit and Centigrade readings. Ans. 9½° F.; — 12½° C.

3. (*a*) How much will an iron tie-rod 60 ft. long expand when the temperature is raised from 40° to 110° ? (*b*) Calculate, also, by formula **70.** (*c*) What is the difference of the two results ?

Ans. (*a*) .345744". (*b*) .345725". (*c*) .000019".

4. To what temperature must a steel tire of 59.93" internal diameter be raised in order that its diameter may be 60.0015" ? Original temperature = 71°. Ans. 270°.

5. What is the volume of .68 lb. of nitrogen gas having a tension of 20 lb. per sq. in. and a temperature of 345° ? Ans. 10.44 cu. ft.

HEAT PROPAGATION.

1117. Heat is propagated through matter and space in three different ways—by *conduction*, by *convection*, and by *radiation*.

1118. Conduction is the slow progress of the vibratory motion from places of higher to places of lower temperature in the *same* body. The rate at which heat is conducted varies greatly with different substances, the *good conductors* being those in which conduction is most rapid, and the *bad* *conductors* being those in which it is very slow. The metals furnish the best conductors, and of these, silver stands first, and copper second. The fluids, both liquid and gaseous, are very poor conductors of heat. Water, for example, can be made to boil at the top of a vessel while a cake of ice is fastened within a few inches of the surface. If thermometers are placed at different depths, while *water boils at the top*, it is found that the conduction of heat downwards is very slight.

1119. Representing the conductivity of silver by 100, the following table shows the conducting power of a number of the metals:

Silver	100.0	Iron	11.9
Copper	73.6	Steel	11.6
Gold	53.2	Lead	8.5
Brass	23.1	Platinum	8.4
Zinc	19.0	Rose's Alloy	2.8
Tin	14.5	Bismuth	1.8

Organic substances conduct heat poorly. This enables trees to withstand great and sudden changes in the atmosphere without injury. The bark is a poorer conductor than the wood beneath it. Cotton, wool, straw, bran, etc., are all poor conductors. Rocks and earth are poorer conductors as the less dense and homogeneous is the mass; hence, the length of time required for the sun's rays to penetrate the earth. The mean highest temperature of the air near the ground in Central Europe is in the month of July, but at a depth of from 25 to 28 feet in the earth it is in the month of December.

1120. Convection is the transfer of heat by the motion of the heated matter itself. It can, therefore, take place only in fluids and gases. For example, as heat is applied to the *bottom* and *sides* of a vessel of water, the densities of the heated particles decrease, and they are crowded up by the heavier ones which take their places. There is thus a constant circulation going on, which tends to equalize the temperature of the whole by bringing the hot portions in contact with the colder, and also greatly facilitates the *conduction* of heat among the *molecules*.

1121. Radiation of heat is the communication of heat from a hot body to a colder one, across an intervening space between them. The best example of radiant heat is that received from the sun, the intervening space in this case being 93,000,000 miles. A person standing in front of a fire, but at some distance from it, feels a sensation of warmth which is not due to the temperature of the air, for, if a screen be interposed between him and the fire, the sensation immediately ceases, which would not be the case if the surrounding air had a high temperature. Hence, bodies can send out rays which excite heat and which penetrate the air without heating it. This, of course, is **radiant heat,** and *takes place in all directions around the body.*

1122. The intensity of heat radiation from a given source

1.—Varies as the temperature of the source.

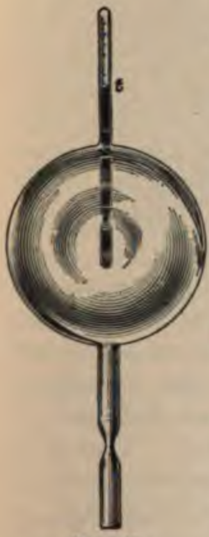

FIG. 217.

2.—Varies inversely as the square of the distance from the source; and,

3.—Grows less as the inclination of the rays to the given radiant surface grows less.

The truth of all these laws has been established by careful experiment, and the second is still further verified by mathematical calculations.

1123. *Radiant heat is transmitted in a vacuum as well as in air.* This is demonstrated by the following experiment:

In the top of a glass flask a thermometer is fixed in such a manner that its bulb occupies the center of the flask. See Fig. 217. The neck of the flask is carefully narrowed by means of the blowpipe; the flask is then attached to an air pump, and a vacuum produced in the interior. This being accomplished, the tube is sealed at the narrow part. On immersing in hot water, or on bringing the flask near some hot charcoal, the mercury is seen to rise at once. It can rise only by reason of the radiation through the vacuum in the interior, for glass is such a poor conductor that the heat could not travel with sufficient rapidity through the sides of the flask and the stem of the thermometer to cause this almost instantaneous rise.

1124. The **radiating power of heated surfaces** also depends very greatly upon the form, shape, and the material of which they are composed. If a cubical vessel, filled with hot water, has one of its vertical sides coated with polished silver, another with tarnished lead, a third with mica, and the fourth with lampblack, experiment has shown that the *radiating power* will be respectively about in the ratio of 2.5:45:80:100, or that bright surfaces radiate less heat than dark ones having the same temperature.

In the same way it is found that the heat-absorbing power of bodies varies in a similar manner. Lampblack *reflects* few of the heat rays which impinge upon it; nearly all are absorbed, while, on the other hand, polished silver reflects the greater part of the radiations, and absorbs only about 2½ per cent.

Some substances neither absorb nor reflect the heat rays to any extent, but transmit nearly all of them just as glass transmits light. For example, rock salt reflects less than 8 per cent. of the radiation it receives, absorbs almost none, and transmits 92 per cent.

1125. It will now be seen that there is a system of exchange going on between heated bodies at all times, which tends to an equalization of temperature. The hot bodies are always cooling, and the cold bodies are always tending towards a rise in temperature, so that heat is created only to be diffused and apparently lost. That it is *not* lost, however, will be shown in the subsequent pages.

1126. Dynamical Theory of Heat.—Before going any farther, it will be convenient to explain here the view now generally taken as to the mode in which heat is propagated.

On this subject, it is stated in Ganot's Physics: "A hot body is one whose molecules are in a state of vibration. The higher the temperature of a body, the more rapid are these vibrations, and a diminution in temperature is but a diminished rapidity of the vibrations of the molecules. The propagation of heat through a bar is due to a gradual communication of this vibratory motion from the heated part to the rest of the bar. A good conductor is one which readily takes up and transmits the vibratory motion from molecule to molecule, while a bad conductor is one which takes up and transmits the motion with difficulty. But, even through the best of the conductors, the propagation of this motion is comparatively slow. How, then, can be explained the instantaneous perception of heat when a screen is removed from a fire, or when a cloud drifts from the face of the sun? In this case, the heat passes from one body to another without affecting the temperature of the medium which transmits it. In order to explain these phenomena, it is imagined that all space, the space between the planets and the stars, as well as the interstices in the hardest crystal and the heaviest metal—in short, matter of any kind—is permeated by a medium having the properties of matter of infinite tenuity, called **ether.** The molecules of a heated body, being in a state of intensely rapid vibration, communicate their motion to the ether around them, throwing it into a system of waves which travel through space and pass from one body to another with the velocity of light. When the undulations of the ether reach a given body, the motion is given up to the molecules of that body, which, in their turn, begin to vibrate; that is, the body becomes heated. This process of this motion through the ether is termed radiation, and what is called a ray of heat is merely one series of waves moving in a given direction."

HEAT MEASUREMENT.

1127. The Unit of Heat.—There are three units in use for measuring the *quantity of heat* given up or absorbed by a body when heated or cooled.

1128. The British Thermal Unit.—*The amount of heat necessary to raise one pound of water one degree Fahrenheit is called a* **British thermal unit.** Instead of writing out the words British thermal unit in full, it is customary to abbreviate them to B. T. U. Thus, 5 pounds of water raised 10° F. would equal 5 × 10 = 50 B. T. U. The unit of heat used here will be the B. T. U.

1129. The Thermal Unit.—*The amount of heat necessary to raise one pound of water 1° C. is called a* **thermal unit.** Since 1° C. = $\frac{9}{5}$ × 1° F., it follows that the thermal unit is $\frac{9}{5}$ times as large as a B. T. U. Hence, to change B. T. U. into thermal units, multiply the number of B. T. U. by $\frac{5}{9}$. To change thermal units into B. T. U., multiply the number of thermal units by $\frac{9}{5}$.

The thermal unit is used by American and British scientific writers, as being better adapted to their calculations.

1130. The Calorie.—*The amount of heat necessary to raise one kilogram of water 1° C. is called a* **calorie.** One kilogram = 2.2 pounds and 1° C. = $\frac{9}{5}$ × 1° F.; hence, a calorie = 2.2 × $\frac{9}{5}$ = 3.96 B. T. U. The calorie is used in France and in other countries using the metric system of weights and measures.

SPECIFIC HEAT.

1131. When equal weights of two different substances, having the same temperature, are placed in similar vessels and subjected for the same length of time to the heat of the same lamp, or are placed at the same distance in front of the same fire, it is found that their final temperature will differ considerably; thus, mercury will be much hotter than water. But as, from the conditions of the experiment, they have each been receiving the same amount of heat, it is clear that the quantity of heat which is sufficient to raise the temperature of mercury through a certain number of degrees will raise the same weight of water through a less number of degrees; in other words, it requires more heat to raise a certain weight of water one degree than it does to raise the same weight of mercury one degree. Conversely, if the same quantities of water and of mercury at 200° be allowed to cool down to the temperature of the room, the water will require a much longer time for the purpose than the mercury; hence, in cooling through the same number of degrees, water gives up more heat than does mercury.

1132. *The number of B. T. U., or parts of a B. T. U., required to raise the temperature of one pound of any substance 1° F. is called the* **specific heat** *of that substance.* It will be seen from the above definition that *the specific heat of a substance is the ratio between the amount of heat required to raise the temperature of the substance 1°, and the amount of heat required to raise the temperature of the same weight of water 1°.*

If the specific heat of lead were given as .0314, it would mean that the amount of heat required to raise a certain weight of lead 1° would raise the same weight of water only .0314 of 1°, or it would mean that .0314 B. T. U. would raise the temperature of one pound of lead 1° F.

EXAMPLE.—The specific heat of copper is .0951; how many B. T. U. will it take to raise the temperature of 75 pounds 180°?

SOLUTION.—Since it takes .0951 B. T. U. to raise 1 lb. of copper 1°, it will take $.0951 \times 75 \times 180$ to raise 75 lb. 180°. Hence, $.0951 \times 75 \times 180 = 1{,}283.85$ B. T. U. Ans.

1133. In the example just given, if it had been required to raise 75 lb. of water 180°—that is, from the freezing point to the boiling point—it would have taken $75 \times 180 = 13{,}500$ B. T. U., and $\frac{1{,}283.85}{13{,}500} = .0951 =$ the specific heat of copper. The following is the formula for finding the number of B. T. U. required to raise the temperature of a substance a given number of degrees, or for finding the number of B. T. U. given up by a body in cooling a given number of degrees:

Let W = weight of body in pounds;

s = specific heat of substance composing the body;

t = original temperature of body;

t_1 = final temperature of body;

n = number of B. T. U. required, or given up, in changing the temperature of the body from $t°$ to $t_1°$.

Then,

$$n = W(t_1 - t)\,s. \qquad \textbf{(72.)}$$

EXAMPLE.—A piece of wrought iron weighing 31.3 lb., and having a temperature of 900°, is cooled to a temperature of 60°; how many units of heat did it give up? The specific heat of wrought iron is .1138.

SOLUTION.—Apply formula **72,** $n = W(t_1 - t)s$. Substituting $n = 31.3\,(900 - 60)\,.1138 = 2{,}992.03$ B. T. U. Ans.

If a body be cooled from a temperature t down to a temperature t_1, the value of n will be negative, the minus sign indicating that the body was cooled.

1134. In the following table are given the specific heats of a number of substances under constant pressure:

TABLE 21.

SOLIDS.

Copper	0.0951	Cast Iron	0.1298
Gold	0.0324	Lead	0.0314
Wrought Iron	0.1138	Platinum	0.0324
Steel (soft)	0.1165	Silver	0.0570
Steel (hard)	0.1175	Tin	0.0562
Zinc	0.0956	Ice	0.5040
Brass	0.0939	Sulphur	0.2026
Glass	0.1937	Charcoal	0.2410

LIQUIDS.

Water	1.0000	Lead (melted)	0.0402
Alcohol	0.7000	Sulphur "	0.2340
Mercury	0.0333	Tin "	0.0637
Benzine	0.4500	Sulphuric Acid	0.3350
Glycerine	0.5550	Oil of Turpentine	0.4260

GASES.

	Constant Pressure.	Constant Volume.
Air..................	0.23751	0.16847
Oxygen............	0.21751	0.15507
Nitrogen............	0.24380	0.17273
Hydrogen...........	3.40900	2.41226
Superheated Steam...	0.48050	0.34600
Carbonic Oxide......	0.24790	0.17580
Carbonic Acid......	0.40400	0.15350

1135. The reason that there are two values for the specific heat of gases is that it takes less heat to raise the temperature of a gas when the volume is constant than when the pressure is constant but the volume varies. Thus, consider a closed cylinder filled with gas. If heat be applied, the pressure and temperature will both increase. Denoting the specific heat for constant pressure by s_p, and for constant volume by s_v, the number of heat units required to heat the gas from $t°$ to $t_1°$ will be $s_v \, W (T_1 - T)$. If, however, the cylinder be imagined to be fitted with a frictionless piston, free to move up or down, and heat be applied, the gas will expand, overcoming a resistance equal to the weight of the piston, plus the pressure of the atmosphere. Hence, in addition to the heat required to increase the vibratory movement of the molecules, heat is also required to overcome the outer pressure which remains constant in this case. The number of heat units necessary will then be $s_p \, W (T_1 - T)$. This subject will be more fully discussed later.

1136. Mixing Two Bodies of Unequal Temperatures.—If a certain quantity of water having a temperature of 40° be mixed with a like quantity having a temperature of 100°, it is evident that the temperature after mixing will be $\frac{40 + 100}{2} = 70°$. But, if 5 lb. of water having a temperature of 40° be mixed with 5 lb. of copper having a temperature of 100°, the temperature after mixing will not be 70°. The resulting temperature may be found by the following formula, provided there is no change of the state in the body (ice melting into water, etc.):

$$t = \frac{W_1 s_1 t_1 + W_2 s_2 t_2 + W_3 s_3 t_3 + etc.}{W_1 s_1 + W_2 s_2 + W_3 s_3 + etc.}, \qquad \textbf{(73.)}$$

in which t is the final temperature of the mixture; W_1, s_1, and t_1, the weight, specific heat, and temperature, respectively, of one body; W_2, s_2 and t_2, the same for second body; and W_3, s_3, and t_3, the same for a third body, etc.

Remembering that the specific heat of water is 1, and getting the specific heat of copper from Table 20, the temperature t will be

$$t = \frac{5 \times 1 \times 40 + 5 \times .0951 \times 100}{5 \times 1 + 5 \times .0951} = 45.21°\text{, nearly.}$$

Example.—If 21 lb. of water at a temperature of 52° is mixed with 40 lb. of water at a temperature of 160°, what is the temperature of the mixture?

Solution.—Since the specific heat of water is 1, it may be left out, in applying formula **73,** and the temperature is found to be

$$t = \frac{21 \times 52 + 40 \times 160}{21 + 40} = 122.82°. \quad \text{Ans.}$$

Example.—A copper vessel weighing 2 lb. is partly filled with water having a temperature of 80° and weighing 7.8 lb. A piece of wrought iron weighing 3 lb. 4 oz. and having a temperature of 780° is dropped into this water. What is the final temperature of the mixture?

Solution.—Substituting the values given in formula **73,** and remembering that the original temperatures of the copper vessel and the water which it contains are the same (3 lb. 4 oz. = 3.25 pounds), we have,

$$t = \frac{2 \times .0951 \times 80 + 7.8 \times 80 + 3.25 \times .1138 \times 780}{2 \times .0951 + 7.8 + 3.25 \times .1138} = 110.97°\text{, nearly.} \quad \text{Ans.}$$

Example.—A wrought-iron ball weighing one pound is placed in a reheating furnace; when it has attained the temperature of the furnace it is taken out and placed in a copper vessel weighing ¼ pound and containing exactly 2 pounds of water at a temperature of 75°. Assuming that no water escapes as steam, and that the temperature of the ball, water, and vessel after mixing is 156°, what is the temperature of the furnace?

Solution.—Substituting the values given in formula **73**,

$$156 = \frac{1 \times .1138 \times t_1 + 2 \times 75 + .5 \times .0951 \times 75}{1 \times .1138 + 2 + .5 \times .0951}, \text{ or}$$

$$156 = \frac{.1138\, t_1 + 153.56625}{2.16135}, \text{ or}$$

$$156 \times 2.16135 = .1138\, t_1 + 153.56625.$$

Hence,

$$.1138\, t_1 = 183.60435, \text{ or}$$

$$t_1 = \frac{183.60435}{.1138} = 1{,}613.4°. \quad \text{Ans.}$$

1137. By means of formula **73**, the specific heat of a substance may be obtained.

Thus, in $t = \dfrac{W_1 s_1 t_1 + W_2 s_2 t_2 + W_3 s_3 t_3 + etc.}{W_1 s_1 + W_2 s_2 + W_3 s_3 + etc.}$,

suppose that the specific heat s_3 be required and all of the other quantities, including t, are known.

Then, solving the above equation for s_3,

$t\,(W_1 s_1 + W_2 s_2 + etc.) + t\, W_3 s_3 = W_1 s_1 t_1 + W_2 s_2 t_2 + W_3 s_3 t_3 + etc.$, or $t\, W_3 s_3 - t_3\, W_3 s_3 = W_1 s_1 t_1 - W_1 s_1 t + W_2 s_2 t_2 - W_2 s_2 t + etc.$,

$$\text{or} \quad s_3 = \frac{W_1 s_1 (t_1 - t) + W_2 s_2 (t_2 - t) + etc.}{W_3 (t - t_3)}. \qquad \textbf{(74.)}$$

Example.—A silver vessel weighing 13 oz. is suspended by a string; 1 lb. 4 oz. of water having a temperature of 120° is poured into it, and in this is placed a piece of metal weighing 14 oz. and having a temperature of 100°. If the temperature of the vessel was 72°, and the temperature of the mixture is 117°, what is the specific heat of the piece of metal?

Solution.—Using formula **74**, and letting W_1, s_1, and t_1 represent, respectively, the weight, specific heat, and temperature of the silver vessel, W_2, s_2, and t_2 ditto for the water, and W_3, s_3, and t_3 the same for the piece of metal,

$$s_3 = \frac{W_1\, s_1 (t_1 - t) + W_2\, s_2 (t_2 - t)}{W_3 (t - t_3)} =$$

$$\frac{13 \times .057 (72 - 117) + 20 \times 1 (120 - 117)}{14 (117 - 100)} =$$

$$\frac{-33.345 + 60}{238} = .112. \quad \text{Ans.}$$

All weights must be reduced to either pounds or ounces before substituting.

EXAMPLES FOR PRACTICE.

1. How many units of heat are required to raise the temperature of 10 oz. of platinum from 80° to 2,000°? Ans. 38.88 B. T. U.

2. In order to determine the specific heat of a certain alloy, a piece weighing 12½ oz. was heated to a temperature of 320°, and was then immersed in 2 lb. 6 oz. of water contained in a lead vessel weighing 4 lb. 7 oz. The temperature of the water and of the vessel being 70°, what was the specific heat of the alloy if the temperature of the mixture was 79°? Ans. .1202.

3. In order to determine the temperature of a chimney, a silver bar weighing 20 oz. is placed in it until it has attained the same temperature. It is then immersed in 1 lb. of water contained in a brass vessel weighing 10 oz. The temperature of the vessel and water being 65°, and of the mixture 98½°, what was the temperature of the chimney? Ans. 596°.

4. An iron casting weighing 3 tons is cooled from 2,100° to 100°; (*a*) how many units of heat does it give up? (*b*) If all this heat could be utilized, how many pounds of coal would it be equivalent to, assuming that 1 lb. of coal gives out 14,500 B. T. U. during its combustion?

Ans. { (*a*) 1,557,600 B. T. U.
(*b*) 107.42 lb.

LATENT HEAT.

1138. In all that has been said in the preceding pages, only the phenomena relating to sensible heat have been considered. If a quantity of pounded ice at a temperature of 32° be put in a vessel and held over the flame of a spirit lamp, heat passes rapidly into the ice and melts it; but a thermometer resting in this mixture of ice and water shows no tendency to rise; it will remain at 32° until all of the ice has been melted. Where has the heat gone that was supplied to the ice? This question was first investigated by Dr. Black, of Edinburgh, in 1760, and is easily explained by the modern dynamical theory of heat.

Dr. Black took a pound of water and a pound of ice, both having a temperature of 32°, and placed them in two vessels suspended in a chamber which was kept at as nearly a uniform temperature as possible. At the end of half an hour the temperature of the water was 39.2°, but the ice did not reach that temperature until 10½ hours had passed, being melted, of course, in the meantime. Dr. Black reasonably assumed that the ice received the same quantity of heat that the water did in each half-hour, because it was placed in exactly the same position in regard to the surrounding air; that is to say, it received $39.2 - 32 = 7.2$ units of heat every half-hour, or 14.4 units every hour, and $14.4 \times 10\frac{1}{2} = 151.2$ units in $10\frac{1}{2}$ hours. Hence, it took $151.2 - 7.2 = 144$ units of heat to change the one pound of ice at 32° into water at 32°; more accurate determinations have fixed this number as 142.65, and this value will be used in this volume whenever the occasion arises for using it. If a pound of water having a temperature of 212° be mixed with a pound of water having a temperature of 32°, the temperature of the mixture will be $\frac{212 + 32}{2} = 122°$, the boiling water giving up 90° and the cold water receiving 90°, thus bringing both to a common temperature. If a pound of ice at a temperature of 32° be mixed with a pound of water at a temperature of 212°, the temperature of the mixture will be only 50.675°, instead of 122, as in the previous case. Here, the water has given up $212 - 50.675 = 161.325$ units of heat in order to bring both bodies to a common temperature. Since the temperature of the ice was raised from 32° to 50.675°, it follows that $50.675 - 32 = 18.675$ units of heat were used to raise the temperature of the ice *after it had been melted* into water, and $161.325 - 18.675 = 142.65$ units of heat were necessary to convert the ice at 32° into water of the same temperature. This extra number of units of heat, which is necessary to convert a solid into a liquid of the same temperature without raising the temperature of the solid, is called the **latent heat of fusion,** and the temperature at which this change of state in the body takes place is called the **melting point,** or **temperature of fusion.** All solids probably have a latent heat of fusion, the word *probably* being used because some solids have never been melted, except at such high temperatures that accurate measurements are not possible, but its value varies greatly for different substances, being greater for ice than for any other known solid, while for frozen mercury its value is only 5.09; that is, to change one pound of frozen mercury at its temperature of fusion (− 37.8°F.) into liquid mercury of the same temperature requires only 5.09 units of heat. Now, it is reasonable to suppose that if it requires 142.65 units of heat to convert a pound of ice at 32° into water at 32°, then the same number of heat units would be given up when water at 32° is changed into ice at 32°; experiment has verified this.

1139. If water be heated to its boiling point of 212° under a constant pressure of 14.69 lb. per sq. in., it has been found by experiment that it will require about 966 units of heat per pound of water to change it into steam at 212°. This extra number of units of heat necessary to convert a liquid into a gas, or, rather, vapor, of the same temperature and pressure is called the **latent heat of vaporization,** and the temperature at which this change of state takes place is called the **temperature of vaporization.**

1140. According to the modern theory of heat, the extra quantity of heat necessary for a change of state of a body is used in forcing the molecules of a body farther apart, and in overcoming the force of cohesion. This latent heat is not lost, but performs work in giving additional energy to the molecules of a body, and it always reappears when the body resumes its former state. Thus, for instance, a pound of steam under a pressure of one atmosphere contains $966 + 180 = 1{,}146$ units of heat more than does a pound of water at 32°. Hence, if 1 lb. of steam at 212° be mixed with $\frac{966}{180} = 5.37$ lb. of water at 32°, the temperature of the mixture will be exactly 212°, or the boiling point of water; in other words, the steam raised 5.37 lb. of water from the freezing point to the boiling point without lowering its own temperature, by merely changing from steam into water. If a pound of water at a temperature of 32° be changed into ice of the same temperature, 142.65 units of heat will be given up during this change of state.

1141. In the following table are given the temperatures of fusion and of vaporization, and the latent heats of fusion and vaporization, whenever they have been determined with sufficient accuracy:

TABLE 22.

Substance.	Temperature of Fusion.	Temperature of Vaporization.	Latent Heat of Fusion.	Latent Heat of Vaporization.
Water	32°	212°	142.65	966.6
Mercury	—37.8°	662°	5.09	157
Sulphur	228.3°	824°	13.26	
Tin	446°	....	25.65	
Lead	626°	....	9.67	
Zinc	680°	1,900°	50.63	493
Alcohol	Unknown	173°	....	372
Oil of Turpentine	14°	313°	....	124
Linseed Oil	....	600°		
Aluminum	1,400°			
Copper	2,100°			
Cast Iron	2,192°	3,300°		
Wrought Iron	2,912°	5,000°		
Steel	2,520°			
Platinum	3,632°			
Iridium	4,892°			

The following example will show the purpose of Tables 21 and 22; their use will be further illustrated in a later section.

EXAMPLE.—How many units of heat will it be necessary to use in changing 12 lb. of ice at a temperature of — 20° C. into steam of 212°?

SOLUTION.—By formula **65**, $t_f = \frac{9}{5} \times -20 + 32 = -4°$ F. This is equivalent to $32° + 4° = 36°$ F. below the freezing point. In Table 21, specific heat of ice was given as .504; hence, it will take $12 \times 36 \times .504 = 217.728$ B. T. U. to raise the temperature of 12 lb. of ice from $-4°$ to 32°. To convert this ice into water of 32° will require $142.65 \times 12 = 1,711.8$ B. T. U. To raise this water from 32° to a temperature of 212° will require $12 \times 180 = 2,160$ B. T. U. To convert it into steam of 212° will require $966.6 \times 12 = 11,599.2$ B. T. U. The total number of units of heat required will be $217.728 + 1,711.8 + 2,160 + 11,599.2 = 15,688.728$ B. T. U. Ans.

1142. A solid may be changed into a liquid, not only by melting it, but also by dissolving it, as salt or sugar is dissolved in water. Since the particles of the solid body must be torn asunder, in opposition to the forces which hold them together, it is reasonable to suppose that a certain amount of heat will be required to do this. That such is a fact may be easily proven by any one having a thermometer. Put a thermometer in a vessel of water, and leave it there until it indicates the temperature of the water, then put in some salt or sugar, and stir so as to make it dissolve more quickly, and it will be found that the mercury has fallen several degrees. In fact, if any solid be dissolved in a liquid that does not act chemically upon it, the temperature of the mixture will be lower than if the solid did not dissolve. It is this principle that is taken advantage of in the so-called freezing mixtures. A mixture of one part of nitrate of ammonia and one part of water will reduce the temperature from 50° to 4°, a fall of 46°. The effects are still more striking when both bodies are solids, one of which is already at the freezing point. Thus, a mixture of two parts of snow, or finely pounded ice, and one part of common salt, will reduce the temperature from 50° to 0°, a range of 50°, while a mixture of four parts of potash and three parts of snow, or pounded ice, will lower the temperature from 32° to – 51°, a range of 83°.

1143. Latent heat plays an important part in everyday life. It takes a long time and severe cold to freeze the water of a river to any depth, even though the thermometer goes far below the freezing point. This is because 142.65 units of heat must be given up by every pound of water, after being brought to the freezing point, before the ice can form. If it were not for this, the rivers, lakes, and other bodies of water would be frozen solid as soon as the water reached the freezing point, and would be melted as soon as the temperature went above that point. In the spring all of the snow on the hills would be melted during a warm day, and great floods would be the consequence. As it is, 142.65

units of heat must be supplied to every pound of snow at 32° to convert it into water at 32°.

EXAMPLE.—How many units of heat will it take to evaporate 25 lb. of mercury from a temperature of 70°?

SOLUTION.—The temperature of vaporization of mercury is 662°, and the specific heat is .0333; the increase in temperature from 70° will be 662° − 70° = 592°. The number of units of heat required will be 25 × 592 × .0333 = 492.84 heat units. The latent heat of vaporization is 157; hence, 492.84 + 25 × 157 = 4,417.84 B. T. U. will be required. Ans.

EXAMPLES FOR PRACTICE.

1. If a pound of steam at 212° and 7 pounds of ice at 32° are mixed what will be the resulting temperature? Ans. 50.5°.

2. How many units of heat are required to vaporize 10 lb. of mercury from a temperature of 100? Ans. 1,757.146 B. T. U.

3. How many pounds of oil of turpentine at 60° can be vaporized by 1 lb. of coal, if the coal gives out 14,500 B. T. U. during combustion? Ans. 62.56 lb.

4. How many pounds of water at 32° can be vaporized by 1 pound of coal? Ans. 12.646 lb.

5. How many pounds of coal are required to raise 100 lb. of wrought iron from 85° to its melting point? Ans. 2.219 lb.

SOURCES OF HEAT AND COLD.

1144. Different Sources of Heat.—Heat is derived from the following sources: 1.—**Physical sources**—that is, the radiation of heat from the sun, terrestrial heat, change of state in bodies and electricity. 2.—**Chemical sources,** or molecular combinations, more especially combustion. 3.—**Mechanical sources,** comprising friction, percussion, and pressure.

1145. Physical Sources.—(1) The most intense of all of the sources of heat is the sun. The majority of scientists are of the opinion that all of the heat received or given up by the earth has, or has had, its source in the sun It would be out of place here to elucidate this theory fully, and the subject will be explained as subdivided above. It is the amount of heat radiated from the sun, and received by the earth, that causes the change of seasons; that causes the water in the rivers, lakes, and seas to evaporate and form the clouds, to be again precipitated as rain or snow. Without it, no living thing, animal or vegetable, could exist.

(2) The earth possesses a heat peculiar to itself, called **terrestrial heat.** When a descent is made below the surface, the temperature is found to gradually increase. This is not caused by the heat radiated from the sun, for the material comprising the earth is such a poor conductor that the heat of the sun's rays penetrates only a very short distance below the surface. The explanation usually given for this phenomenon is that the interior of the earth is in a molten condition. The terrestrial heat exerts but a slight effect, not raising the temperature of the surface more than $\frac{1}{20}$ of a degree.

(3) If a liquid be poured upon a finely divided solid, as a sponge, flour, starch, roots, etc., the temperature will be increased from 1° to 10°, according to conditions. This phenomenon might be called *heat produced by capillarity.*

(4) The heat produced by a change of state has already been described; it is the heat given off when a body is converted from a gas or liquid to a liquid or solid.

(5) Extremely high temperatures may be produced by the electric current. By means of it, quick-lime, firebrick, osmium, porcelain, and several other substances, which, until very recently, have resisted every attempt to melt them, may be made to run like water.

1146. Chemical Sources.—Whenever two or more substances which act chemically upon one another are brought together and allowed to combine, heat is evolved. When this phenomenon is produced by oxygen uniting with carbon, or other substance, and is accompanied by light, it is called **combustion.** This subject will not be treated of here; it is usually considered by itself in works on chemistry, steam boilers, and metallurgy.

1147. Mechanical Sources.—(1) The friction between any two bodies rubbed together produces heat. Rubbing one hand briskly against the other will soon make the hands too warm for comfort. The friction between a journal and its bearing causes heat; the heat causes the journal and bearing to expand, the journal expanding more rapidly on account of being smaller and being heated more quickly; the expansion causes a greater pressure on the bearing, producing more friction and heat. If the bearing is not properly oiled, the heat will become so intense in a short time that the soft metal in the bearings will melt. When shooting stars strike the earth's atmosphere their velocity is so great (sometimes as high as 150 miles a second) that the friction of the atmosphere causes them to take fire almost instantly. Wherever there is friction, there is heat.

(2) Heat is also generated by percussion. The repeated blows of a hammer upon a piece of iron, lead, or other metal, will soon make it quite hot.

(3) The generation of heat by pressure was spoken of in connection with gases—that is, the temperature rises when a gas is compressed. This is also true of solids and liquids, but the results are not so marked in their cases. The production of heat by the compression of gases is easily shown by means of the *pneumatic syringe* shown in Fig. 218. This consists of a glass tube with thick sides, hermetically closed with a leather piston. At the bottom is a small cavity in which a piece of cotton, moistened with ether or carbon disulphide, is placed. The tube being filled with air, the piston is suddenly plunged downwards. Thus compressed, the air generates so much heat that the cotton is ignited, which can be seen to burn when the piston is suddenly withdrawn. The

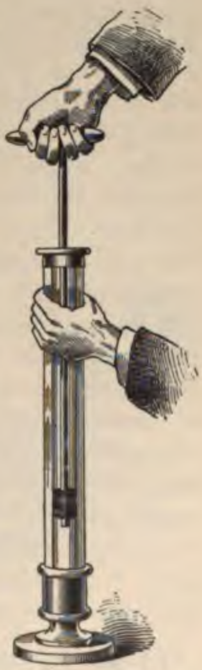

FIG. 218.

ignition of the cotton in this experiment indicates a temperature of at least 570°, since it will not ignite at a lower temperature.

THE PRODUCTION OF MECHANICAL WORK BY HEAT.

THE MECHANICAL EQUIVALENT OF HEAT.

1148. From what has been previously stated, it should now be evident that heat is a kind of energy, since, when a body is heated, the heat imparted to it manifests itself in giving the molecules a greater velocity, and in forcing them farther apart, in opposition to the force of cohesion, which tends to draw them together and reduce their velocity; but this requires energy, and heat is the form of energy used. Again, when a body is cooled, heat is given up; in other words, the energy of the molecules is lessened. The heat thus given up can be used to heat or impart energy to other bodies. Since heat is a kind of energy, it is reasonable to suppose that there is some relation between energy (or work) and heat. From careful experiment it has been found that one unit of heat (1 B. T. U.) is equivalent to 778 foot-pounds of work—that is, 778 foot-pounds of work would be required, to be expended by friction or otherwise, to raise a pound of water 1° in temperature under a pressure of one atmosphere, and that the heat given up by 1 pound of water in cooling 1°, if used as energy, could raise 1 pound to a height of 778 feet, or 778 pounds 1 foot. This number, 778, has been obtained in many ways, and is called **the mechanical equivalent of heat.** It is denoted in all formulas into which it enters, in books treating of heat, by the letter J, the initial letter in the name of Dr. Joule who first determined its value with any degree of accuracy.

1149. The First Law of Thermodynamics.—*Heat is energy, and has capacity for doing work; the number of units of work which can thus be performed by a given* *quantity of heat is proportional to the number of units of heat in that quantity.* This law is more concisely stated as follows: *Heat and mechanical energy are mutually convertible.*

For example, if a weight of 778 pounds be dragged 20 feet along a horizontal surface, and the coefficient of friction between the weight and the surface is .25, the work alone will be $778 \times 20 \times .25 = 3{,}890$ foot-pounds. If this had been done in such a manner that the entire movement could have taken place in water, say an upright shaft turning in a pivot bearing, the friction thus produced could raise the temperature 1° of $\frac{3{,}890}{778} = 5$ lb. of water; or 1 lb., 5°; or 10 lb., $\frac{1}{2}^\circ$, etc. Here, the mechanical energy necessary to overcome the friction was converted into heat. The amount of heat obtainable from a given amount of mechanical energy is always the same, and is in the proportion of one British thermal unit to every 778 foot-pounds of work.

1150. A fine illustration of the conversion of mechanical energy into heat is given by the experiment shown in Fig. 219. A brass tube, about 7 in. in length and $\frac{3}{4}$ in. in diameter, is attached to a small wheel, by means of a cord passing around this wheel and a larger one turned by a handle, as shown; the tube is three-fourths full of water, and is closed with a cork. The tube being held by the clamp

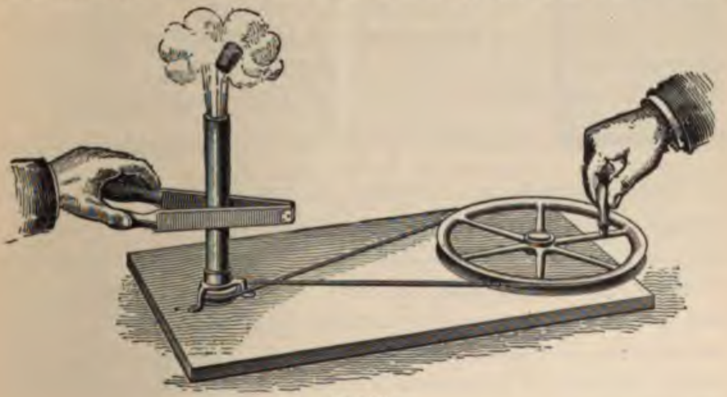

FIG. 219.

and made to rotate rapidly by means of the larger wheel, considerable friction is generated, causing the water within the tube to be heated; the temperature rapidly increases, and part of the water is converted into steam, whose pressure becomes so great as to force out the cork. Suppose that the weight of the water is 1½ oz., that its original temperature was 60°, and that a pressure of 10 lb. per sq. in. was necessary to force out the cork. From the steam and water tables, to be given in connection with the subject of Steam and Steam Engines, the number of heat units in a pound of water above a temperature of 32° from which steam of 10 lb. pressure is being given off is 209.39, and the number of heat units in a pound of steam of this pressure above 32° is 1,155.1. To create this pressure, it can be shown that the weight of the steam in the tube above the water will be about .0005 oz.; hence, the weight of the water will be $1.5 - .0005 = 1.4995$ oz. Since 1 oz. $= \frac{1}{16}$ of a pound, the number of heat units in the water will be $\frac{1.4995}{16} \times 209.39 = 19.6238$. The number of heat units in the steam will be $\frac{.0005}{16} \times 1{,}155.1 = .0361$, nearly. The sum is $19.6238 + .0361 = 19.6599$. From the steam tables above referred to, the number of heat units in a pound of water (above 32°) having a temperature of 60°, and under a pressure of one atmosphere, is 28.00626; in 1½ oz. there would be $\frac{1.5}{16} \times 28.00626 = 2.6256$ heat units. Consequently, the number of heat units necessary to be supplied in order to blow out the cork is $19.6559 - 2.6256 = 17.0303$. The theoretical number of foot-pounds of work which would have to be exerted in turning the large wheel to accomplish this result would be $17.0303 \times 778 = 13{,}249.57$ foot-pounds. The actual amount of heat used would be greater than that just calculated, for the reason that some is lost by radiation and conduction, and in overcoming the friction of the bearings.

Suppose that enough heat were lost to bring the total number of foot-pounds of work up to 15,000, and that it took 10 minutes to cause the cork to be blown out; the number of foot-pounds per minute would be $\frac{15,000}{10} = 1,500$ and the horsepower exerted would be $\frac{1,500}{33,000} = \frac{1}{22}$ H. P.

1151. Having shown that mechanical work can be changed into heat, it will now be demonstrated that heat can be changed into mechanical work. Fig. 220 represents a cylinder *A B* partly filled with gas or air confined within the cylinder by means of the piston *P*. The gas is then under a pressure of the atmosphere, and has also an additional pressure due to the weight of the piston. If heat be applied to the bottom of the cylinder, the piston will gradually rise in proportion to the amount of heat supplied. In expanding, it will have to do work in order to raise the piston. Suppose a rope, fastened to the piston and passed over a pulley, to have a weight on the other end a trifle less than the total pressure of the atmosphere plus the weight of the piston. Now, if the gas within the cylinder be cooled, the piston will fall, owing to the combined weight of the piston and the pressure of the atmosphere, and raise the weight, thus performing work. In the first case, a certain amount of heat was supplied to the gas to do work; in the second case, heat was *taken away* from the gas (cooled) in order that work might be done. In both cases the amount of work done was proportional to the amount of heat supplied or taken away, and, had the work done been the same, the amount of heat supplied or taken away would also have been the same.

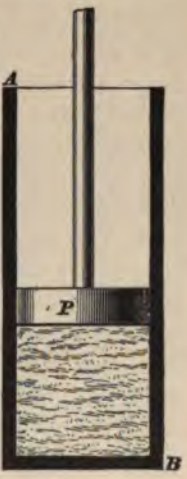

FIG. 220.

1152. When a body free to expand is heated, two operations are performed: 1. The temperature is raised and its volume is increased. 2. The body, in expanding, overcomes the outer pressure, and thus does work. The coefficient of cubic expansion of mercury is .0001001, or say .0001. Suppose that 1 cubic foot of mercury be confined in a non-expanding vessel, having a diameter corresponding to a circle whose area is 1 sq. ft. The height of the mercurial column will then be 1 ft. Let the mercury be heated until its temperature is 100° higher than before; the volume will be increased $.0001 \times 100 = .01$ cu. ft. Since the area could become no larger (being confined in a non-expanding vessel), the column of mercury must be .01 ft. longer than it was before being heated. In expanding, the pressure of the atmosphere (equaling a weight of $144 \times 14.7 = 2{,}116.8$ lb.) was overcome through this distance, and work was done equivalent to $2{,}116.8 \times .01 = 21.168$ foot-pounds. The greater part of the heat went to increase the temperature, and to push the molecules farther apart against the force of cohesion tending to pull them together. This is called the **inner work.** The work of overcoming the outside pressure through a certain distance, by expanding, is called the **outer work.** The outer work for the above case was found to be 21.168 foot-pounds; the inner work may be found as follows: The specific heat of mercury under constant pressure, taken from Table 21, is .0333; hence, to raise the temperature of 1 cu. ft. (= 850 lb.) 100° will require $850 \times 100 \times .0333 = 2{,}830.5$ heat units, equivalent to a total work of $2{,}830.5 \times 778 = 2{,}202{,}129$ foot-pounds. Subtracting the outer work, the inner work equals $2{,}202{,}129 - 21.168 = 2{,}202{,}107.832$ foot-pounds. In the case of a solid body, this difference would be still more marked. In fact, the outer work is so slight, compared with the inner work in solid and liquid bodies, that it is usually neglected, except in the case of water.

1153. In the case of gases, however, the outer work plays a very important part, as a little consideration will show. Thus, suppose that air was substituted for the mercury in the previous case, and was prevented from escaping from the vessel by a piston without weight, as shown in Fig. 221. Let the original temperature of the air be 70°, and let it be heated until the temperature is 100° higher, or 170°. The new volume is determined by formula **58,** Art. **1054,** to be $V_1 = \frac{460 + 170}{460 + 70} = 1.19$ cu. ft., nearly. Hence, the increase $= 1.19 - 1 = .19$ cu. ft., and the piston is raised .19 ft. The outer work will be $2,116.8 \times .19 = 402.192$ foot-pounds. The weight of a cu. ft. of air having a temperature of 70° is found by formula **71,** Art. **1116,** $p V = W R T$, or $14.7 \times 1 = W \times .37052 \times 530$, or

$$W = \frac{14.7}{.37052 \times 530} = .07486 \text{ lb., nearly.}$$

The specific heat of air for constant pressure is .23751; hence, the total number of heat units required is $.07486 \times 100 \times .23751 = 1.778$ heat units. $1.778 \times 778 = 1,383.284$ foot-pounds. Since the outer work required 402.192 foot-pounds, the inner work will require $1,383.284 - 402.192 = 981.092$ foot-pounds. This shows that, in the case of air and gases, the outer work is a little less than half the inner work. Since the force of cohesion has no perceptible effect in the case of gases, the inner work tends only to raise the temperature, or, in other words, to increase the vibratory movement of the molecules. Consequently, if the piston in Fig. 221 were fastened down, so that the volume of the gas would remain the same, there would be no outer work, and the total work required to raise the temperature 100° would be 981.092 foot-pounds, or to raise the temperature 1°, 9.81092 foot-pounds. The inner work may also be calculated by using the specific heat for constant volume as directed in Art. **1135.** Thus, inner work $= s_v W (T_1 - T) \times 778 = .16847 \times .07486 \times 100 \times 778 = 981.18$ foot-pounds. The slight difference in results is due to decimals.

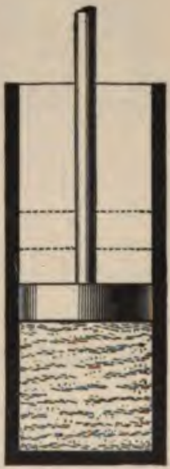

FIG. 221.

1154. Some bodies do not always expand when heat is applied, as, for example, water. When *water* of 32° is heated to 33°, its volume decreases, and this continues until a temperature of 39.2° has been reached; beyond this point, the volume increases with the temperature, and, consequently, the density decreases, since when the weight remains the same, and the volume grows larger, the density decreases. The reason of this apparent contradiction to the heat theory is that the increase in density is caused by the pressure of the atmosphere; the heat added to the water until the temperature of 39.2° is reached gives a freer movement to the molecules, and enables the atmospheric pressure to force them nearer together. If it were not for this pressure, it would take more heat to raise the temperature of a pound of water 1° than now; this is evident from analogy to the cylinder and piston in Fig. 221; when the piston was fastened down, it took far less heat to increase the temperature of the air than when it was free to move. In the case of water, the specific heat varies somewhat with the pressure, while in the case of gases it is practically constant. In consequence of this variation, the latent heat of ice is diminished by heavy pressures—that is, its melting point is lowered. Under a pressure of 13,000 atmospheres (191,100 lb. per sq. in.), ice will melt at 0° instead of at 32°.

WORK DONE BY EXPANSION OF AIR AND GASES.

1155. Isothermal Expansion. — When a gas expands, it does work; when it is compressed, work is required to be done upon the gas to compress it. Suppose that a certain quantity of air is confined in a vessel having an area of 1 sq. ft., and whose length is 5 ft., plus the thickness of the piston, so that the piston can move 5 ft. Suppose the piston to be in the position shown in Fig. 222, and that the absolute pressure of the volume of air enclosed in the cylinder is 100 lb. per sq. in. on the piston, and that the temperature is 150°. Since the area of the piston is 1 sq. ft., the volume of the enclosed air is 1 cu. ft. Now, let this air expand, and keep the temperature constant by adding heat to it. The piston will move ahead; the atmospheric pressure upon it will be overcome through the distance it moves; the volume of the air will increase and the pressure decrease, according to Mariotte's law. When the piston has moved 1 ft., the volume will be 2 cu. ft., and the pressure is found by the formula $p_1 v_1 = p_2 v_2$, to be $100 \times 1 = p_2 \times 2$, or $p_2 = 50$ lb. per sq. in. When the piston has moved 2 ft., the pressure is $\frac{100}{3} = 33\frac{1}{3}$lb. per sq. in., etc.

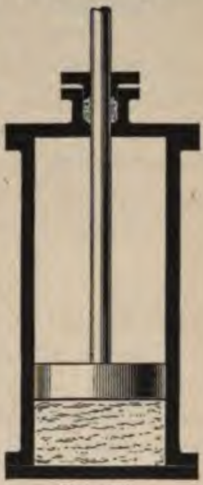

FIG. 222.

1156. To show the effects of this expansion upon the pressure and volume graphically, two indefinite straight lines are drawn at right angles to each other, as *O Y* and *O X*, in Fig. 223. Any line drawn from *O X* parallel to *O Y* is called an **ordinate.** Choose a convenient scale, say 1 in. = 1 cu. ft., and lay off *O L* = 1 in. = 1 ft. of cylinder length = 1 cu. ft. of cylinder volume = the volume of air admitted at full pressure before expanding. Make *O F* = 5 in. = the total travel of the piston = the total volume after the piston has reached the end of the cylinder. Now, choose another scale to represent the pressures, say 1 in. = 20 lb. The length of a line representing 100 lb. would be $\frac{100}{20} = 5$ in. Lay off this distance on *O Y*, thus locating the point *H*. The pressure is 100 lb. per sq. in. throughout the distance *O L*; hence, drawing *H M* parallel to *O X*, it is evident that any ordinate measured from *O X* to this line, with a scale of 1″ = 20 lb., will equal 100 lb. pressure per sq. in. When the piston reaches the point *L*, no more air is admitted, and as it begins to move away from the position *A L*, the pressure begins to fall, the volume increasing in proportion to the distance of the piston from *O Y*. The pressures corresponding to a number of different positions of the piston, calculated by the formula $p v = p_1 v_1 = p_2 v_2$ etc., are as follows:

When piston has moved $\frac{1}{2}$ ft., or to d, pressure = $66\frac{2}{3}$ lb
" " " 1 " K, " = 50 "
" " " $1\frac{1}{2}$ " f, " = 40 "
" " " 2 " I, " = $33\frac{1}{3}$ "
" " " $2\frac{1}{2}$ " h, " = $28\frac{4}{7}$ "
" " " 3 " G, " = 25 "
" " " $3\frac{1}{2}$ " k, " = $22\frac{2}{9}$ "
" " " 4 " F, " = 20 "

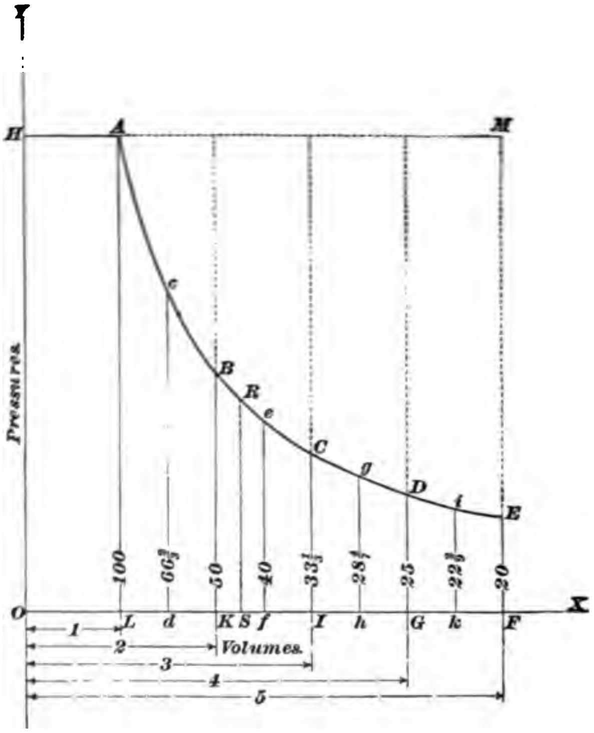

FIG. 223.

At the points d, K, f, I, h, G, k, F, erect ordinates, and make them equal in length to the pressure at that point, to the scale of 1 in. = 20 lb.—that is, make $c\,d = 66\frac{2}{3}$ lb., $B\,K =$ 50 lb., etc., and through the points A, c, B, e, C, g, D, i, E, draw the curve shown in this figure. If care has been taken in drawing this figure, any ordinate drawn from a point on the line $O\,X$, and limited by the curve, will indicate the pressure of the air in the cylinder when the piston is at that point. Thus, suppose it is desired to know the pressure when the piston is at the point S. Erect the ordinate $S\,R$, and measure it with the same scale that was used to draw the curve; the reading on the scale will be the pressure at that point.

1157. In order to find the work done by the air while the piston was traveling from L to F, and during which time the pressure fell from $A\,L$, or 100 lb. per sq. in., to $E\,F$, or 20 lb. per sq. in., the average pressure, or **mean ordinate,** must be known. This can be found by dividing the area of $A\,E\,F\,L$ by its length $L\,F$. That this statement may be clearly understood, suppose a semicircle to be drawn as shown in Fig. 224, having a diameter of 6 in. Its area will be $6^2 \times .7854 \div 2 =$ 14.1372 sq. in. Divide this area by the length, 6 in., thus $14.1372 \div 6 = 2.3562$ in. On the diameter as a side, and with 2.3562 in. for another side, construct the rectangle $A\ B\ C\ D$; the area of this rectangle will evidently be the same as the area of the semicircle.

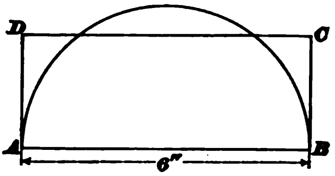

FIG. 224.

1158. **Rule.**—*No matter what the shape of an area may be, if any line be drawn through it and limited by lines perpendicular to it, and tangent to the bounding line of the area, the product of the length of this line, and the mean ordinate drawn from this line to the bounding line, will be equal to the area of that part included by the line, the tangents, and the bounding line included between the points of tangency.*

Thus, in Fig. 225, if the length of $A\,B$ is known, and $B\,D$, perpendicular to it, is tangent to the bounding line $A\,E\,D\,I$, and $E\,F$ is the mean perpendicular (mean ordinate) from $A\,B$ to the bounding line $A\,E\,D\,B$, $A\,B \times E\,F =$ the area of $A\,E\,D\,B\,A$; if $G\,H$ is the mean ordinate from $A\,C$ to $A\,H\,C$ of the area $A\,H\,J\,C$, $A\,C \times G\,H =$ area of $A\,H\,J\,C$, and if $L\,K$ is the mean ordinate from $I\,B$ to $I\,D\,B$, $I\,B \times L\,K =$ the area $I\,L\,D\,B$. Conversely, if an area is given, and the length of a line in it, so located that perpendiculars to the extremities of the line are tangent to the bounding line of the area, the mean ordinate to this line equals the area divided by the length of the line. Thus, the mean ordinate of $A\,E\,D\,B = \frac{\text{area of } A\,E\,D\,B}{A\,B} = E\,F$, etc. Returning now to Fig. 223, if the area $A\,E\,F\,L$ is known, the mean ordinate (mean pressure) can be found by dividing this area by the length $L\,F$.

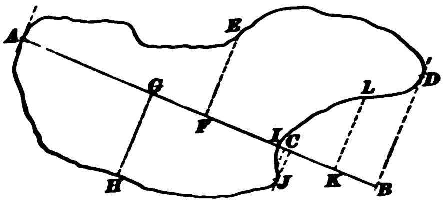

FIG. 225.

1159. The area may be found in two ways: 1. Approximately, by dividing the figure into a number of small areas, adding the ordinates at the center of each of these small areas, and dividing the sum by the number of areas; this result, multiplied by the length $L\,F$, is the area $A\,E\,F\,L$. 2. Exactly, by using the planimeter, an instrument for measuring plane areas.

The first method, as applied to Fig. 223, is shown in Fig. 226. $L\,F$ is divided into 8 equal parts, and ordinates are erected at the points of division, thus dividing the area $A\,E\,F\,L$ into 8 small areas. At the *middle points* of these areas, the ordinates *1-1*, *2-2*, *3-3*, etc., are drawn and measured, the lengths (measured to the same scale used to lay off $A\,L$) being marked on the drawing. The sum of these ordinates is $80 + 57.1 + 44.4 + 36.4 + 30.8 + 26.7 + 23.5 + 21 = 319.9$ lb. Hence, the mean pressure $= 319.9 \div 8 =$

39.99 lb. per sq. in. The second method, by using the planimeter, will not be described, since instructions always go with the instrument. Calling the mean pressure 40 lb. per sq. in., the work which the air could do in expanding from L to F at a constant temperature would be equal to the area of the piston in square inches, multiplied by the mean pressure per sq. in., multiplied by the distance through which it moves or works = 144 × 40 × 4 = 23,040 foot-pounds.

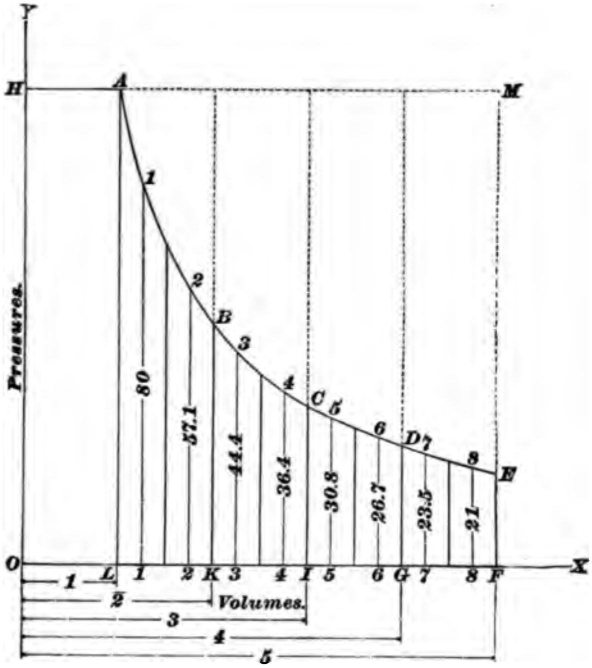

FIG. 226.

1160. The work can also be calculated directly, without constructing the diagram, by means of the following formula, in which

L = the work in foot-pounds;
P = the total initial pressure in pounds per square foot;
P_1 = the total final pressure in pounds per square foot;
V = the initial volume in cubic feet;
V_1 = the final volume in cubic feet.

$$L = 2.3026\, PV \log \frac{V_1}{V}. \qquad \textbf{(75.)}$$

Since $PV = P_1 V_1$, $\frac{P}{P_1} = \frac{V_1}{V}$ and formula **75** might be written

$$L = 2.3026\, PV \log \frac{P}{P_1}. \qquad \textbf{(76.)}$$

Whichever formula is used, it must be kept in mind that the fraction $\frac{V_1}{V}$ or $\frac{P}{P_1}$ must *always be greater than 1—that is, the numerator must always be greater than the denominator.*

Substituting the values used in Fig. 223, formula **75** or **76** gives

$$L = 2.3026 \times (144 \times 100) \times 1 \times \log \frac{5}{1} =$$
$$2.3026 \times 144 \times 100 \times .69897 = 23{,}176 \text{ foot-pounds, nearly.}$$

This is the actual value, and shows that the approximate method used in the previous work was very close.

Suppose that the number of parts had been doubled—that is, that the line LF had been divided into 16 equal parts, instead of 8—the sum of the ordinates drawn at the middle of these parts would then have been

$88.9 + 72.7 + 61.5 + 53.3 + 47.1 + 42.1 + 38.1 + 34.8 + 32 + 29.6 + 27.6 + 25.8 + 24.2 + 22.9 + 21.6 + 20.5 = 642.7$. $642.7 \div 16 = 40.17$ lb. per sq. in. $144 \times 40.17 \times 4 = 23{,}138$ foot-pounds, nearly.

Where a table of logarithms is not at hand, a sufficiently close result for all practical purposes can be obtained by dividing $AEFL$ into 10 parts.

1161. The curve shown in Fig. 223 is called the **isothermal expansion curve,** or the **expansion curve of constant temperature.** It is known in mathematics as the **equilateral hyperbola,** and hence, when used on indicator diagrams, is sometimes called the **hyperbolic expansion curve.** If the initial volume, pressure, and final volume are known, the curve may be constructed graphically without calculating the different points, as was done in Fig. 223. Thus, in Fig. 227, let *O Y* and *O X* be two lines at right angles to each other. These lines are known in mathematics as the **coordinate axes,** the line *O Y* being called the **axis of ordinates,** or **axis of *Y*,** and the line *O X*, the **axis of abscissas,** or **axis of *X*.** Let *O A* represent the absolute initial pressure, and *O B* the initial volume. Through *A* draw the indefinite straight line *A M* parallel to the axis of *X*, and through *B* draw the indefinite straight line *B F* parallel to the axis of *Y*. The point *C*, where these two lines meet, is the point where the expansion is to begin; consequently, it is one point on the curve. Through the point *O*, called the **origin,** and which is point of no volume and no pressure, draw a number of lines, *O F*, *O D*, *O N*, *O M*, etc., cutting *B F* at *F*, *1*, *5*, etc., and *A M* at *4*, *D*, *N*, etc. Through the points *F*, *1*, *5*, etc., draw lines parallel to the axis of *X*, and through *4*, *D*,

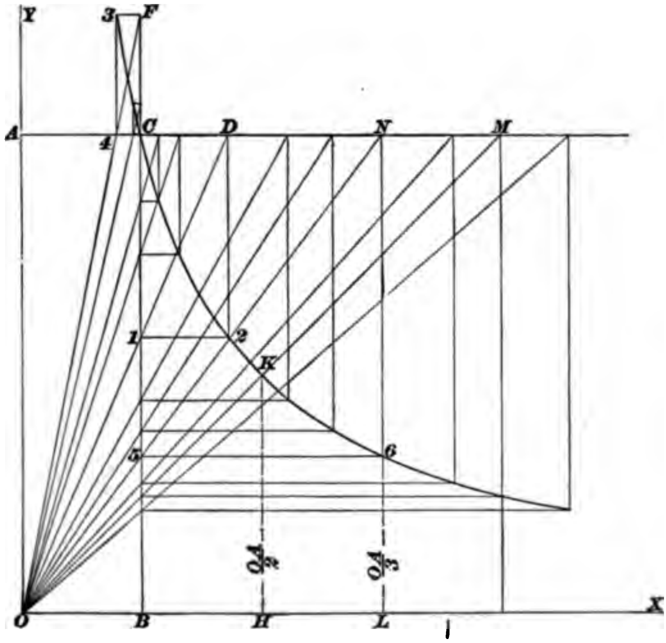

FIG. 227.

N, etc., draw lines parallel to the axis of *Y*. These lines intersect in the points *3*, *2*, *6*, etc., which are points on the required isothermal expansion line. To prove this, lay off *B H* equal to *O B*, and draw *H K* parallel to the axis of *Y*, intersecting the curve in *K*. Now, if *K* is a point on the isothermal expansion line, *H K* must be equal in length to one-half of *O A*, since, when the volume is twice as great, the pressure is only half as great. Similarly, if $H L = B H = O B$, *L 6* must be one-third as long as *O A*. By measurement this will be found to be the case. This curve and method of constructing it is much used in "working up" indicator diagrams, and will be further explained in connection with the subject of Steam and Steam Engines.

1162. If the air or gas be compressed, the action will be exactly the reverse of the expansion. Heat would have to be abstracted instead of added; the pressure would increase instead of decreasing, and the volume decrease instead of increasing.

In Fig. 228, which is Fig. 223 repeated, let *E F* represent the initial pressure = 20 lb. per sq. in., *O F* the initial volume = 5 cu. ft. As the volume decreases, the pressure will increase, as indicated by the isothermal curve *E D C B A*, when the temperature is kept constant. The curve may be constructed as shown in Fig. 227, by taking *O* as the point from which to draw the lines *O D*, *O N*, etc., in Fig. 227. The point *F* could not be taken from which to draw these lines, for they must always be drawn from the point of no pressure and no volume. *F* is a point of no pressure, but it indicates a volume of 5 cu. ft. The work required to compress the air under these conditions may be calculated by formula **75** or **76**, remembering that the larger volume or pressure must be in the numerator of the fraction.

Formulas **75** and **76** then become

$$L = 2.3026\,P V \log \frac{V}{V_1} \qquad \textbf{(77.)}$$

$$\text{and } L = 2.3026\,P V \log \frac{P_1}{P}, \qquad \textbf{(78.)}$$

in which the letters have the same meaning as before.

1163. Formulas **75, 76, 77,** and **78** will be easier to use if the pressure be taken in pounds per square inch, and $144 \times 2.3026 = 331.5744$ be substituted for 2.3026. As before, the volume must always be taken in cubic feet. Formulas **75** and **76** then become

$$L = 331.5744\, p\, V \log \frac{V_1}{V} \qquad \textbf{(79.)}$$

and $$L = 331.5744\, p\, V \log \frac{p}{p_1}, \qquad \textbf{(80.)}$$

in which p = pressure in pounds per square inch.

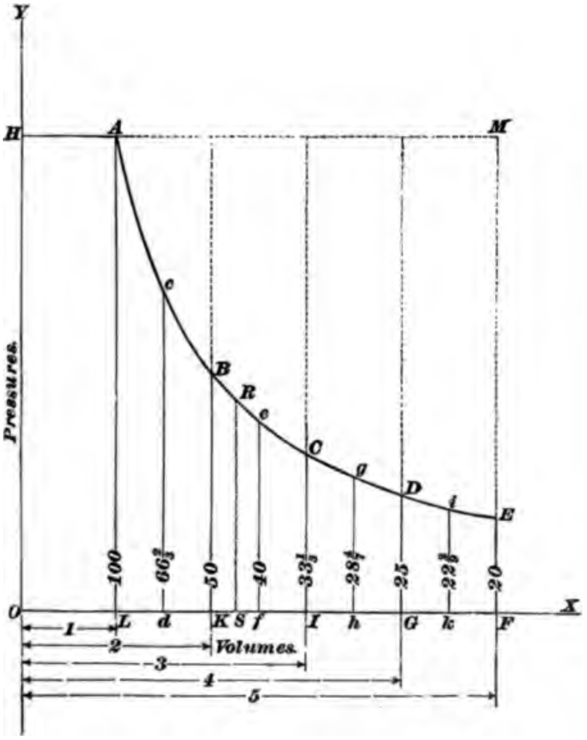

FIG. 228.

EXAMPLE.—The initial volume of a body of gas which is to be compressed is 6.78 cu. ft. The initial pressure is 16 lb. per sq. in. If this gas be compressed until its tension is 144 lb. per sq. in., what work will have to be expended, the temperature being kept constant?

SOLUTION.—Using formula **80**, and remembering that the greater pressure must be in the numerator,

$$L = 331.5744\, p\, V \log \frac{p_1}{p} = 331.5744 \times 16 \times 6.78 \times \log \frac{144}{16} = 331.5744$$

$$\times\, 16 \times 6.78 \times .95424 = 34{,}323.24 \text{ foot-pounds.} \quad \text{Ans.}$$

1164. Adiabatic Expansion.—Suppose that a volume of air expands from the same initial volume and pressure as in the case of Fig. 223, but that no heat is added or taken away. The temperature will fall during expansion, and rise during compression. The pressure will fall much faster than in the case of isothermal expansion, and increase much faster than in isothermal compression for the same increase or decrease in volume. The air expands no longer, according to the law $p\,v = p_1\,v_1 = p_2\,v_2$, etc., but according to another law which can only be proven by the use of higher mathematics; this law is for air:

$$p\,v^{1.41} = p_1\,v_1^{\,1.41} = p_2\,v_2^{\,1.41}, \text{ etc.} \qquad \textbf{(81.)}$$

In other words, the pressure multiplied by the 1.41 power of the corresponding volume is a constant, and is equal to the product of the pressure and 1.41 power of the volume at any other part of the stroke. The initial volume is 1 cu. ft., and the initial pressure is 100 lb. per sq. in. in Fig. 223. Using these values in the present case, $p\,v^{1.41} = 100 \times 1^{1.41} = 100$; hence, $p_1\,v_1^{\,1.41} = 100$, $p_2\,v_2^{\,1.41} = 100$, etc.

Assuming the different volumes, the pressures may be calculated as follows:

Let $v = 2\tfrac{1}{2}$ cu. ft.; then, $p \times 2.5^{1.41} = 100$, or $p = \dfrac{100}{2.5^{1.41}}$;

$$\log p = \log 100 - 1.41 \log 2.5 =$$

$$2 - 1.41 \times .39794 = 2 - .56110 = 1.43890,$$

$$\text{or } p = 27.47 \text{ lb., nearly}$$

Calculating in this manner the pressures corresponding to the different values of the volumes for points corresponding to the volume points in Fig. 223, the following results are obtained:

Pressure corresponding to volume $c\,d = 56.5$ lb.
" " " $B\,K = 37.63$ "
" " " $e\,f = 27.47$ "
" " " $C\,I = 21.25$ "
" " " $g\,h = 17.1$ "
" " " $D\,G = 14.6$ "
" " " $i\,k = 12.-$ "
" " " $E\,F = 10.34$ "

Making the different ordinates equal in length to these pressures, and using the same scale as before—1 in. = 20 lb.

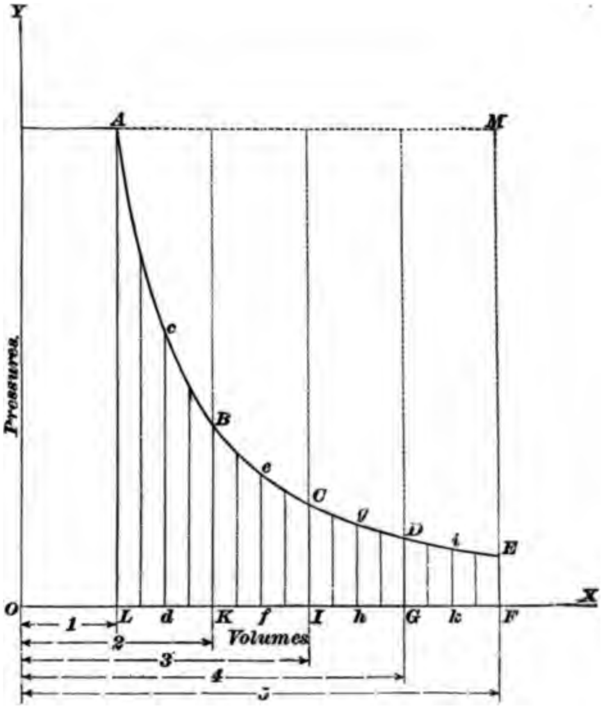

FIG. 229.

—the curve shown in Fig. 229 is produced by tracing the line $A\ B\ C\ D\ E$ through these points. It will be noticed

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that the area of $AEFL$ in Fig. 229 is considerably smaller than in Fig. 223; consequently, the mean pressure is less, and the work done in expanding is less. This was to be expected, since, no heat being added, the temperature must fall and with it the pressure also. Erecting ordinates at the middle points of these divisions, and measuring them in a manner similar to the approximate method of finding the mean pressure followed in Fig. 226, the mean pressure is found to be

$$\frac{73 + 45.5 + 31.9 + 24 + 19 + 15.5 + 13 + 11.1}{8} = 29\tfrac{1}{8} \text{ lb. per sq. in.}$$

The work done is evidently $144 \times 29\tfrac{1}{8} \times 4 = 16,776$ foot-pounds.

1165. The mathematical formula which gives the work directly when the initial and final volumes and the initial pressure are known is

$$L = 2.44\,PV\left[1 - \left(\frac{V}{V_1}\right)^{.41}\right]. \qquad (82.)$$

By means of formula **82**, just given, the work is found to be 16,974 foot-pounds. Thus, substituting the values given and remembering that P = pressure in pounds per square foot,

$$L = 2.44 \times (100 \times 144) \times 1\left[1 - \left(\frac{1}{5}\right)^{.41}\right] = 2.44 \times 14,400$$
$$(1 - .51691) = 16,974 \text{ foot-pounds.}$$

1166. If the initial and final pressures and the initial volume are given, to find the work a formula may be derived as follows:

From formula **81**, $PV^{1.41} = P_1 V_1^{1.41}$; hence, $\frac{V^{1.41}}{V_1^{1.41}} = \frac{P_1}{P}$, or $\frac{V}{V_1} = \left(\frac{P_1}{P}\right)^{\frac{1}{1.41}}$. Effecting both sides of this last equation with an exponent of .41, there results $\left(\frac{V}{V_1}\right)^{.41} = \left(\frac{P_1}{P}\right)^{\frac{.41}{1.41}}$, or $\left(\frac{V}{V_1}\right)^{.41} = \left(\frac{P_1}{P}\right)^{.29078}$, since $.41 \div 1.41 = .29078$.

Substituting the right-hand member of the last equation in formula **82**,

$$L = 2.44\,PV\left[1 - \left(\frac{P_1}{P}\right)^{.29078}\right]. \qquad \textbf{(83.)}$$

1167. If the pressure be taken in pounds per square inch, **82** and **83** become

$$L = 351.36\,p\,V\left[1 - \left(\frac{V}{V_1}\right)^{.41}\right], \qquad \textbf{(84.)}$$

$$\text{and } L = 351.36\,p\,V\left[1 - \left(\frac{p_1}{p}\right)^{.29078}\right]. \qquad \textbf{(85.)}$$

In both formulas, p and V are the initial pressure and volume, respectively. When a gas expands without receiving or losing any heat, the pressure falls, as shown by Fig. 229, and it is said to expand **adiabatically.** The curved line $A\ B\ C\ D\ E$ is called the **adiabatic curve.**

1168. Formulas **82** and **83** (and, of course, **84** and **85**) may be used for compression as well as for expansion, the letters P and V representing the initial pressure and volume, and P_1 and V_1 the final pressure and volume, in both cases. To show that such is the case, proceed as follows:

Dividing both sides of formula **82** by 2.44,

$$\frac{L}{2.44} = PV\left[1 - \left(\frac{V}{V_1}\right)^{.41}\right].$$

Now, if the formula is true for both cases, the work done during adiabatic expansion must equal the work required to immediately compress the air back to its pressure before expansion. But, if the final pressure and volume after expansion be represented by P_1 and V_1, these letters will represent the initial pressure and volume during compression. Consequently, in order to prove that the formula holds good for both cases, it must be proved that

$PV\left[1 - \left(\frac{V}{V_1}\right)^{.41}\right] = P_1V_1\left[1 - \left(\frac{V_1}{V}\right)^{.41}\right]$. By formula **81**, $PV^{1.41} = P_1V_1^{1.41}$, or $P_1 = \frac{PV^{1.41}}{V_1^{1.41}}$. Representing the exponent 1.41 by m, for convenience, $P_1 = \frac{PV^m}{V_1^m}$. Substituting this value of P_1 in $P_1 V_1 \left[1-\left(\frac{V_1}{V}\right)^{.41}\right]$, it becomes, since $.41 = m - 1$,

$$\frac{PV^m}{V_1^m} \times V_1\left[1-\left(\frac{V_1}{V}\right)^{m-1}\right] = PV \times \frac{V^{m-1}}{V_1^{m-1}}\left[1-\frac{V_1^{m-1}}{V^{m-1}}\right] =$$

$$PV\left[\frac{V^{m-1}}{V_1^{m-1}}-1\right] = -PV\left[1-\frac{V^{m-1}}{V_1^{m-1}}\right] = -PV\left[1-\left(\frac{V}{V_1}\right)^{.41}\right],$$

which was to be proved.

When applying the formula for cases of compression, it will be found that the result is negative. The minus sign merely indicates compression, the numerical value being the same as in the case of expansion.

1169. From formula **81** two other formulas may be derived, which are of great importance in investigations pertaining to the theory of heat. They are derived as follows:

$$\text{Since } PV^{1.41} = P_1 V_1^{1.41},\ \left(\frac{V}{V_1}\right)^{1.41} = \frac{P_1}{P}.$$

Representing 1.41 by m, as before,

$$\left(\frac{V}{V_1}\right)^m = \frac{P_1}{P},\ \text{or } \frac{V}{V_1} = \left(\frac{P_1}{P}\right)^{\frac{1}{m}}. \quad (a)$$

Multiplying both sides of equation (a) by $\frac{P}{P_1}$,

$$\frac{PV}{P_1V_1} = \frac{P}{P_1}\left(\frac{P_1}{P}\right)^{\frac{1}{m}} = \frac{P}{P_1} \times \frac{P_1^{\frac{1}{m}}}{P^{\frac{1}{m}}} = \frac{P_1^{\frac{1}{m}-1}}{P^{\frac{1}{m}-1}} = \left(\frac{P_1}{P}\right)^{\frac{1-m}{m}}.$$

Substituting for m its value, $1 - 1.41 = -.41$, and

$\frac{PV}{P_1V_1} = \left(\frac{P_1}{P}\right)^{-\frac{.41}{1.41}} = \left(\frac{P_1}{P}\right)^{-.29078}$. According to the theory of exponents, see Arts. **564** and **565,** $\left(\frac{P_1}{P}\right)^{-.29078} = \left(\frac{P}{P_1}\right)^{.29078}$

Hence, $\frac{PV}{P_1V_1} = \left(\frac{P}{P_1}\right)^{.29078}$. ($b$)

1170. According to formula **62,** Art. **1058,**

$$\frac{PV}{T} = \frac{P_1V_1}{T_1},\ \text{or } \frac{PV}{P_1V_1} = \frac{T}{T_1}.$$

Substituting this value of $\frac{PV}{P_1 V_1}$ in equation (*b*)

$$\left(\frac{P}{P_1}\right)^{.29078} = \frac{T}{T_1}. \qquad (86.)$$

Likewise, since $.29078 = \frac{1.41 - 1}{1.41} = \frac{m - 1}{m}$, formula **86** may be written $\left(\frac{P}{P_1}\right)^{\frac{m-1}{m}} = \frac{T}{T_1}$. (*c*)

But, since $\frac{P}{P_1} = \left(\frac{V_1}{V}\right)^m$, from formula **81**, equation (*c*) may be written $\left[\left(\frac{V_1}{V}\right)^m\right]^{\frac{m-1}{m}} = \frac{T}{T_1}$, or $\left(\frac{V_1}{V}\right)^{m-1} = \frac{T}{T_1}$.

Substituting for m its value,

$$\left(\frac{V_1}{V}\right)^{.41} = \frac{T}{T_1}. \qquad (87.)$$

1171. Formulas **86** and **87** may be used to compute the temperature of the air after adiabatic expansion or compression when the initial and final pressure or the initial and final volume are known and the initial temperature has been noted.

In formulas **86** and **87**, the pressures, volumes, and temperatures may be expressed in any units desired, remembering that the pressures and temperatures are *absolute*. In other words, the pressures may be in pounds per square inch, pounds per square foot, inches of mercury, etc.; the volumes may be in cubic feet, cubic inches, cubic meters, etc., and the temperatures may be in Fahrenheit, Centigrade, or Reaumur degrees.

EXAMPLE.—The temperature of the air as it enters the cylinder of an air compressor is 60°; what is its final temperature after being compressed to 100 pounds per square inch, absolute, the compression being adiabatic?

SOLUTION.—The initial pressure is, of course, 14.7 lb. per sq. in.; hence, substituting in formula **86** the values of P, P_1, and T, $T_1 = T\left(\frac{P_1}{P}\right)^{.29078} = 520\left(\frac{100}{14.7}\right)^{.29078} = 908.1°$. Therefore, final temperature = 908.1 − 460 = 448.1°. Ans.

1172. If the volume of air were 5 cu. ft. and the pressure were 10.34 lb. per sq. in.—that is, if the piston were at *E F*, Fig. 229, and the air were compressed to 1 cu. ft., no heat being lost—the final pressure would be 100 lb., as before; the curve of pressures would be the adiabatic curve *E D C B A*, as in the case of expansion. The work which the air would do when it expanded isothermally, or at constant temperature, was found to be 23,176 foot-pounds, and when it expanded adiabatically, 16,974 foot-pounds, a result considerably less. This was to be expected, since, as no heat was added, the heat required to do the work of expansion had to be taken from the gas, thus reducing its energy and the amount of work that it could do. To better show the effects of isothermal and adiabatic expansion, the two curves shown in Figs. 223 and 229 are drawn together in Fig. 230. Here *S A B C D E* is the isothermal curve of

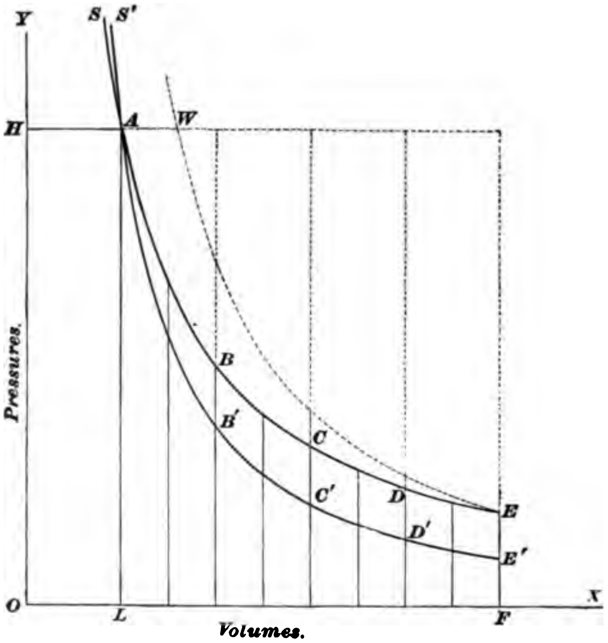


FIG. 230.

expansion or compression, and $S' A B' C' D' E'$ is the corresponding adiabatic curve. If 5 cu. ft. of air having a tension of 20 lb. per sq. in. be compressed isothermally, the curve of compression would follow the line $E D C B A S$, while, if compressed adiabatically, the initial tension and volume being the same, it would follow the dotted line $E W$. Hence, if the air were thus compressed to 1 cu. ft., it is easy to see that the work required would be far more for adiabatic compression than for isothermal compression. To obtain the area $A C' E' F L$, the following formula may be used, which gives it directly for air when p and p_1 are the greater and lesser pressures, respectively, and V and V_1 their corresponding volumes:

$$\frac{p V - p_1 V_1}{.41} = \text{area.} \qquad \textbf{(88.)}$$

1173. By means of this formula, the mean ordinate may be calculated directly, without drawing the curve and measuring the mean ordinates of the equal parts. Thus, the pressure corresponding to a volume of 5 cu. ft., and denoted by the ordinate $E' F$, was found to be 10.34 lb. per sq. in. The greater pressure was 100 lb. per sq. in., and the corresponding volume 1 cu. ft.; hence, the area $A B' C' D' E' F L A$ is $\frac{p V - p_1 V_1}{.41} = \frac{100 \times 1 - 10.34 \times 5}{.41} = 117.805$ sq. in. This, divided by the length $L F = 4$, gives $\frac{117.805}{4} = 29.45125$ lb. per sq. in. = mean ordinate. Since the area of the piston was 144 sq. in., and the piston moved 4 ft., the work it could do is $29.45125 \times 144 \times 4 = 16{,}964$ foot-pounds. The previous calculation gave 16,974 foot-pounds, a difference of 10 foot-pounds. Both methods would have given the same result had the calculation for the final pressure, 10.34 lb. per sq. in., been carried out to a sufficient number of decimal places, and 7-figure logarithms used instead of those of 5 figures. The difference is so slight that the results are practically the same.

1174. A little thought will show that the work done is directly proportional to the areas, and that the areas themselves may be considered as representing the work done on the piston during one stroke. For the mean pressure was just now found to be 29.45 lb. per sq. in. Since every inch of length on any ordinate in Fig. 230 represents a pressure of 20 lb. per sq. in., the actual length in inches of the mean ordinate is $29.45 \div 20 = 1.4725$ in. The length of the area is 4 in., and the actual area is $1.4725 \times 4 = 5.89$ sq. in. Now, since the ordinates are so drawn that 1 in. = 20 lb. pressure per sq. in., and the area of one sq. ft. is 144 sq. in., $5.89 \times 20 \times 144 =$ work $= 16,963$ foot-pounds, the same result as before. Therefore, if in any diagram of this kind the actual area be multiplied by the vertical scale of pressures in pounds per square inch (in this case, 1 in. = 20 lb. per sq. in.) and by the horizontal scale of volumes in cu. ft. (in this case, 1 in. = 1 cu. ft.), and then multiplied by 144, the result is the work. The work is represented by the area, and the ratio of any two areas is the same as the ratio of the works.

1175. A study of the curves *E D C B A* and *E W*, in Fig. 230, will show why the walls of air compressors are cooled. Suppose that *E F* represents a pressure of 14.7 lb. per sq. in., instead of 20 lb., as formerly. This is the pressure of the atmosphere, and, consequently, the initial pressure in the air compressor cylinder. If the air were not cooled while being compressed, the pressures corresponding to the various volumes would be given by the dotted adiabatic curve *E W*.

If the air thus compressed could be used at once, there would be no loss, since the heat imparted to it would be utilized in doing work, and it would make no difference whether the compression was adiabatic or isothermal. Such is not the case, however. The air, after leaving the compressor, is stored in a large reservoir called a **receiver,** from which it is conveyed in pipes to the engines, pumps, or other machines which it operates. These are situated sometimes 5 miles or more from the compressor, and when the air reaches them, its temperature has been reduced to that of the atmosphere. As a consequence of this reduction in temperature, the pressure falls to a point determined by the intersection of an ordinate drawn through the point *W* and the isothermal curve *E B A*. Fig. 231 shows the curves when applied to an air compressor. *O L* represents the initial volume, say 5 cu. ft.; *L B* the atmospheric pressure 14.7 pounds per square inch, and *O H* the final pressure, say 100 pounds per square inch. *B C* represents the adiabatic, and *B A* the isothermal, compression curve, respectively.

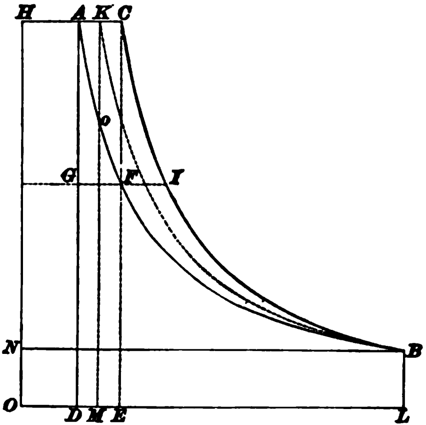


FIG. 231.

The point *F*, where the ordinate through *C* intersects the isothermal *B A*, indicates the pressure of air, when compressed adiabatically after it has cooled to the temperature of the outside air. Measuring the ordinate *E F*, the pressure at the point is found to be 57.26 pounds per square inch.

The work done in compressing the air adiabatically, and in forcing it out of the cylinder is proportional to the area *B L O H C*, while, for isothermal compression, the work is proportional to the area *B L O H A*. The work lost through adiabatic compression is the difference of these two areas, or the area *A B C*. By the use of some cooling device, such as the water jacket described in Art. **1071**, the compression curve will lie between the curves *B C* and *B A*, and the subsequent fall of pressure due to the cooling will be greatly reduced. In the best types of modern air compressors, this curve will lie about half way between *B C* and *B A*, as shown by the dotted curve *B K*, and the fall of pressure will then be Ko, instead of CF, as in the former case where no cooling took place.

1176. The *efficiency* of the cooling device is determined as follows: Suppose that $BKHN$ represents an actual indicator diagram taken from the air cylinder of an air compressor. Lay off NO equal to the pressure of the atmosphere, as determined by a barometer, or equal to 14.7 pounds per square inch, if no barometer reading has been taken, and draw the isothermal and adiabatic curves BA and BC in the usual manner. Then, the efficiency of the cooling device $= \frac{\text{area of } CBK}{\text{area of } CBA}$.

The student who approaches the subject of cooling devices for air compressors for the first time is apt to reason fallaciously in the following manner: He argues that, although the air is cooled, the work done on the air is the same in either case, the work not shown by the card being turned into heat and carried away by the cooling water. The fallacy of this reasoning may be proved by taking an indicator diagram from the steam cylinder. It will always be found that the work shown by the steam diagram will always equal that shown by the air diagram, plus the work needed to overcome the friction of the moving parts, no more and no less. The student may reason himself out of the fallacy, thus: During the compression of a given weight of air, there are four quantities which are liable to vary: the pressure P, the volume V, the temperature T, and the total quantity of heat Q which the air possesses. During adiabatic compression, the total quantity of heat in the air remains the same; i. e., Q is constant while P, V, and T vary. During isothermal compression, on the contrary, P, V, and Q vary, T remaining constant. If Q represents the total amount of heat in the gas before compression, and Q_1 the total amount of heat after compression, $Q - Q_1 = 0$, in the case of adiabatic compression, while, in the case of isothermal compression, $778\,(Q - Q_1)$ is exactly equal to the work represented by the area ABC in Fig. 231.

EXAMPLES FOR PRACTICE.

1. If 5.68 cu. ft. of air having a temperature of 50° is compressed adiabatically to a volume of 1.3 cu. ft., what is the final temperature?
Ans. 473.56°.

2. In the above example, if the initial tension is 14.7 lb. per sq. in., what is the final tension? Ans. 117.57 lb. per sq. in.

3. With the same data as above, calculate the horsepower required to discharge the air, if 10% is allowed for friction and the compressor discharges 30 times per minute? Ans. 46.84 H. P.

4. With the conditions the same as in the preceding example, calculate the horsepower required when the compression is isothermal?
Ans. 30.053 H. P.

5. Eight-tenths cu. ft. of air, at a temperature of 120° and a pressure of 45 lb. per sq. in., expands adiabatically to the pressure of the atmosphere. (*a*) What is the final volume? (*b*) The final temperature? (*c*) The work done during expansion?

$$\text{Ans.}\begin{cases}(a)\ 1.769\text{ cu. ft.}\\(b)\quad -41.07°.\\(c)\ 3{,}513\text{ ft.-lb.}\end{cases}$$

THE IDEAL HEAT ENGINE.

1177. Second Law of Thermodynamics.—*Heat cannot pass from a cold to a hot body by a self-acting process unaided by external agency.*

1178. The Reversible Cycle Process.—In Fig. 232, suppose $S\ S$ to be the cylinder of a single-acting engine; i. e., one which does work only when the piston is moving in one direction, and, for simplicity, assume that the engine is a hot-air engine. Call the fire which heats the air, or *source of heat*, the **hot body**; the atmosphere into which the hot air exhausts, and which absorbs the heat, the *refrigerator*, or **cold body**, and the air in the cylinder which does the work, owing to the expansion, the **intermediate body**. Suppose, further, that the cylinder is made of a perfect non-conducting heat material and that the head (which call *a*) can be removed and replaced, whenever it is desired, by one that is a perfect conductor of heat. Call this head *b*. All of the above conditions regarding the construction of the cylinder are, of course, impossible; the only reason for making these assumptions is that the action of the intermediate body may be considered under perfect conditions.

Let $O\ Y$ and $O\ X$, Fig. 232, be the coordinate axes, and let $O\ V_1$ represent the volume $S\ 1$ of the air in the cylinder, whose absolute temperature is T_1; pressure, P_1, and volume, V_1. When the piston is at *1*, the line $V_1 A$ represents the pressure P in pounds per square foot.

1. Suppose the head *b* to be in place, and to be in contact with the hot body, which is always kept at a uniform temperature T_1, any heat abstracted being immediately supplied by the fire. Then, so long as the head *b* is in contact with the hot body, the temperature of the air in the cylinder will remain constant. Suppose the air to expand until the piston has reached another position, as *2*, overcoming a resistance at every point just equal to the tension of the expanding air. Heat is supplied by the hot body and the temperature remains constant; in other words, the expansion is isothermal. The work done will be represented by the area $A\ B\ V_2\ V_1$.

2. Replace head *b* with head *a*, and let the air expand further until the piston has reached the extreme position *3*. No heat can now enter or leave the cylinder, and this expansion will be adiabatic. The position *2* should be so chosen that, at the end of the adiabatic expansion, the temperature T_2, corresponding to the pressure $C\ V_3$ and volume $O\ V_3$, which denote by P_2 and V_2, respectively, will be the same as that of the cold body. The work done during this period is represented by the area $B\ C\ V_3\ V_2$, and the total work done during expansion from *1* to *3* by $A\ B\ V_2\ V_1 + B\ C\ V_3\ V_2 = A\ B\ C\ V_3\ V_1$

3. Replace the head *a* with head *b*, and, supposing head *b* to be in contact with the cold body, move the piston to position *4*. The air will then be compressed, and, since the temperature of the cold body is assumed to remain at T_2, the compression is isothermal. Consequently, a certain quantity of heat must be abstracted from the air and rejected to the cold body. The work done *upon* the air will be represented by the area $C\ V_3\ V_4\ D$. The position *4* should be so chosen that if the air be compressed adiabatically from *4* to *1*, the volume, pressure, and temperature of the air, when

the piston reaches position *1*, will be the same as at the beginning.

4. Replace the head *b* with head *a*. No heat can then enter or leave the cylinder, and the air will be compressed adiabatically to the original volume, pressure and temperature, provided the position *4* has been rightly chosen. The work done *upon* the air is represented by the area $D V_4 V_1 A$, and the total work done *upon* the air is $C V_3 V_4 D + D V_4 V_1 A = C V_3 V_1 A D$.

The excess of work done by the air over that done upon it; i. e., the excess of heat in foot-pounds, taken from the hot body over that rejected to the cold body, is determined by difference of the areas $A B C V_3 V_1$ and $C V_3 V_1 A D$, or $A B C D$. It should be noted that the only means by which the piston could be returned from *3* to *1* was through the application of an *external force*. It will also be noticed that the condition of the intermediate body is now exactly the same as regards pressure, volume, and temperature as in the beginning.

1179. A series of operations similar to that described above is called a **cycle process**, and when the last operation leaves the intermediate body in the same state as in the beginning, the process is called a **closed cycle;** otherwise, it is an *open* cycle. Thus, the process represented by the lines $A B$, $B C$, $C D$, and $D A$ is a *closed cycle*, while that represented by the lines $A B$, $B C$, $C F$, and $F E$ is an *open cycle*, and heat must be added to the intermediate body to bring it into the same conditions that governed it in the beginning.

1180. Every closed cycle process is **reversible** ; that is, the operations described in connection with it may be reversed. Thus, let the air expand adiabatically from *1* to *4*, the pressures being represented by the curve $A D$; then, isothermally from *4* to *3*, the pressures following the curve $D C$; then, compress it adiabatically from *3* to *2*, the pressures following the curve $C B$, and, lastly, compress it isothermally from *2* to *1*, the pressures following the curve $B A$.

During the third operation, work is done upon the air (intermediate body), and heat is abstracted by the cold body equal to this work in foot-pounds. The amount of this work is $2.3026\ c\ T_2 \log \frac{V_3}{V_4} = 2.3026\ c\ T_2 \log r_3 =$ area $C V_3 V_4 D$. During the fourth operation, the temperature is raised, owing to the conversion of work upon the air into heat, and the amount of this work is

$2.44\,c\,T_2\left[1 - \left(\frac{V_1}{V_4}\right)^{.41}\right] = 2.44\,c\,T_2\left(1 - r_4^{.41}\right) =$ area $D V_4 V_1 A$. It was shown during the demonstration of the determination of the points B and D that $\frac{V_2}{V_3} = \frac{V_1}{V_4}$. Hence, $r_2 = r_4$, and the work done during adiabatic expansion, or the area $B C V_3 V_2$, equals the work done during adiabatic compression, or the area $D V_4 V_1 A$. Since, in the first case, heat in the intermediate body is converted into work, and, in the second case, work from some external source is converted into heat, the two works, being equal, neutralize each other, and the total work done by the machine and represented by the area $A B C D A$ equals the difference of the work done by the intermediate body during *isothermal expansion* over that done during *isothermal compression*; i. e., work done $= 2.3026\,c\,T_1 \log r_1 - 2.3026\,c\,T_2 \log r_3$.

Consequently, $r_1 = r_3$, and the work accomplished during the cycle $= 2.3026\,c\,T_1 \log r_1 - 2.3026\,c\,T_2 \log r_3 = 2.3026\,c \log r_1 (T_1 - T_2)$. Hence, the efficiency of a perfect heat engine $= \frac{2.3026\,c \log r_1 (T_1 - T_2)}{2.3026\,c\,T_1 \log r_1} = E = \frac{T_1 - T_2}{T_1}$. **(89.)**

That is, *for a perfect heat engine operating through a reversible cycle process, the efficiency of the machine is the ratio of the difference of the absolute temperatures of the sources of heat and of cold to the absolute temperature of the source of heat.*

Since, according to the first law, heat and mechanical energy are mutually convertible, it follows that the fraction $\frac{T_1 - T_2}{T_1}$ represents the percentage of the heat taken from the hot body, which was utilized in doing work.

1183. The efficiency of the engine can become equal to unity, or 100%—i. e., the engine can turn the whole of the heat supplied to it into work—only when $T_2 = 0$, and this can only occur when the cold body has the absolute zero of temperature. The absolute temperature T_2 can not be made 0, nor even approached; in fact, it is impracticable to reduce the temperature below that of the surrounding air; hence, in order to obtain a comparatively high efficiency, the initial temperature must be very high. Suppose that the temperature of the air at the beginning of expansion was 540°, and at the beginning of adiabatic compression was 32°; the absolute temperatures would be $540 + 460 = 1000°$, and 492°, respectively. The efficiency would be $\frac{T_1 - T_2}{T_1} = \frac{1000 - 492}{1000} = 50.8\%$. Such a high temperature could not be used in actual practice. In a practical working engine, the efficiency would be even less than that indicated by the fraction $\frac{T_1 - T_2}{T_1}$, since work is required to overcome the loss due to friction, a part of the heat supplied is radiated, etc. The terms heat and work are here considered to be synonymous.

1184. It is easy to see that a closed cycle is more efficient than an open cycle. For, referring to Fig. 232, let *A B C F E A* represent an open cycle. Then, the work done by the air is the area $A\ B\ C\ V_2\ V_1$ as before, while the work done upon the air when the point *E* has been reached is $C\ V_2\ V_1\ E\ F$. The gain in area over that obtained in the closed cycle is the area *A D F E.* But in order that the temperature of the intermediate body may be the same as that of the hot body, an amount of heat must be imparted equal to the work represented by the area *E G Y A*, and, since this area is evidently greater than the area *A D F E*, it follows that there is a loss over that of the previous cycle.

1185. It is now easy to prove that an engine operating through a cycle between the temperatures T_1 and T_2 can not have a greater efficiency than $\frac{T_1 - T_2}{T_1}$. For, suppose that an engine could be devised having a greater efficiency than the one operating, as indicated by Fig. 232 (which call No. 1); call this engine No. 2, and let it drive No. 1 through a reverse cycle. (For example, suppose engine No. 2 to be a hot-air engine, and No. 1 an air compressor.) Then, engine No. 2 takes heat from the hot body and rejects it into the cold body, while engine No. 1, operating in a reverse cycle, takes heat from the cold body and rejects it into the hot body. Suppose the horsepower of both engines to be the

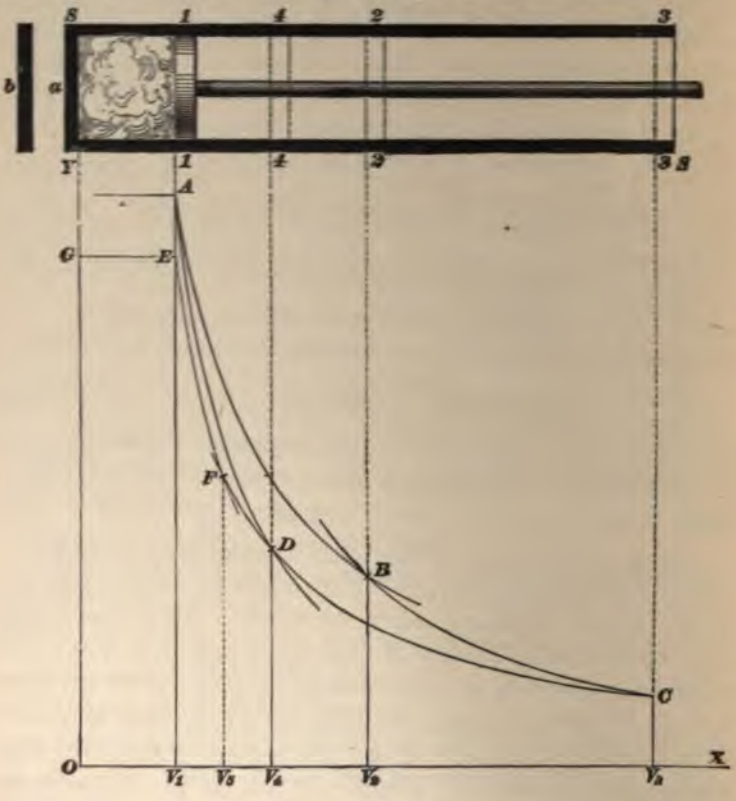


FIG. 232.

same. Then, in engine No. 2, work is done *by* the intermediate body by aid of the heat received from the hot body; and, in engine No. 1, work is done ***upon*** the intermediate body by aid of the heat taken from the cold body through the agency of engine No. 2. If friction be neglected and both engines are perfect engines, it is evident that this combination could go on running forever.

Since the power of both engines is the same, and engine No. 2 was assumed to be more efficient than engine No. 1, it is evident that engine No. 2 will reject less heat into the cold body than engine No. 1 takes from it. From this, it follows that if the engines be kept to work long enough, the whole of the heat in the cold body could be taken out of it and transferred to the hot body—that is to say, heat could be transferred from a cold body to a hot body by means of a self-acting contrivance—a result contrary to all experience, and contradicting the second law of thermodynamics. It is easy to see that the result would be a perpetual motion machine—an impossibility.

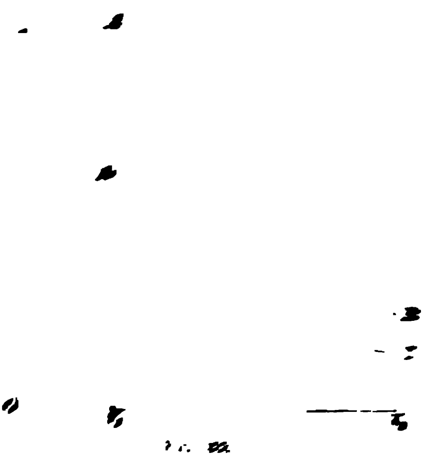
1186. The conclusion is thus evident: *No heat engine operating between the temperatures* T_1 *and* T_2 *can have a greater efficiency than the reversible cycle engine.* Likewise, *the thermal efficiency of any heat engine may be determined by the fraction* $\frac{T_1 - T_2}{T_1}$, *where* T_1 *is the highest and* T_2 *the lowest absolute temperatures of the intermediate body.*

If the student is not satisfied by the above reasoning that no engine can have a greater efficiency than $\frac{T_1 - T_2}{T_1}$, he may assume the intermediate body to be subjected to any process whatever; then, calculate the work done by it and the work done upon it. If between the same limits of temperature he can obtain a greater amount of work for the same quantity of heat taken from the hot body, then the above reasoning is not true.

1187. It was previously shown that a closed cycle had a greater efficiency than an open one. Now, take a cycle

[illegible]

[illegible]



[illegible] would be zero. Hence, in order to obtain useful work from the machine, it is necessary to lower the pressure and, in so doing, the temperature as well. The pressure [illegible] is evidently [illegible] = 35 pounds per square inch. Suppose it be lowered to the pressure of the atmosphere, 14.7 pounds per square inch. This may be done in two ways: either by removing a portion of the [illegible] reducing its weight, or by [illegible] heat. Suppose, for convenience, [illegible] employed. Then the resulting temperature [illegible] formula **59, Art. 1055,** $T_2 = \frac{14.7 \times [illegible]}{[illegible]}$ [illegible], corresponding to a thermometer temperature of 60°. Now, compressing it isothermally, it will follow the curve C D, and the pressure corresponding to a volume of 1 cubic foot will be $\frac{14.7 \times 4}{1} = 58.8$ pounds per square inch. The work done upon the air is 331.5744 × 14.7

$\times 4 \times \log\frac{4}{1} = 11,738$ foot-pounds. The heat energy required to raise the temperature and pressure to the original temperature and pressure is $778 s_v W (T_1 - T_2)$. The weight of 1 cubic foot of air at the temperature T_2 of 520° and a pressure of 58.8 pounds per square inch is, by formula **61**, Art. **1057**, $W = \frac{58.8}{.37052 \times 520} = .3052$ pound, nearly. Hence, the heat energy required $= 778 \times .16847 \times .3052\ (885 - 520) = 14,601$ foot-pounds. Hence, the work accomplished during the cycle $ABCDA = 19,963 - 11,738 = 8,225$ foot-pounds, while the heat energy expended was $19,963 + 14,601 = 34,564$ foot-pounds. Consequently, the efficiency $= \frac{8,225}{34,564} = 23.79\%$. Had the engine operated through a reversible cycle, the efficiency would have been $\frac{885 - 520}{885} = 41.24\%$.

Since a similar result may be obtained for any process which the student may apply the reasoning to, it follows that, under the theoretical conditions governing the reversible cycle process, the reversible cycle is the most efficient.

The foregoing description of the ideal heat engine, and conclusions derived from the consideration of it, comprise the most important laws and generalizations to be found in the science of thermodynamics. The student should study it with extreme care, and review it after finishing the subject of Steam and Steam Engines.

1188. NOTE.—The following application of the foregoing principles to the indicator diagram of a steam engine should not be read until the subject of Steam and Steam Engines has been studied.

In Fig. 234, $B\ C\ D\ E\ F\ G$ represents a diagram taken from a perfect steam engine; i. e., an engine which admits steam at full boiler pressure, cuts off instantly, exhausts at the end of the stroke, the pressure falling instantly to that of the atmosphere, has no back pressure, exhaust closes instantly at the proper point, and which neither radiates nor absorbs heat from the cylinder walls. Since the isothermal of saturated steam is a straight line parallel to the atmospheric line $I\,E$, the clearance, or initial, volume $O\,V_1$ may be regarded as if filled with a mixture of steam and water having the absolute pressure $O\,A$, say 100 pounds per square inch, and the temperature 327.625° corresponding to that pressure. Now, let the piston move to the position $C\,V_2$, the volume increasing to $O\,V_2$, and all the water turning into steam. By addition of heat, the temperature (consequently, the pressure) may be kept constant, and $B\,C$ is the isothermal curve. At C, the supply of heat is stopped and the adiabatic expansion begins, continuing to the end of the stroke, or until the point D is reached. Here the exhaust-valve opens, and the greater part of the steam is allowed to escape into the atmosphere or into the condenser; suppose, for convenience, that it escapes into the atmosphere. The pressure immediately falls to $V_3\,E$. The engine now reverses its stroke and pushes the steam out of the cylinder at the constant pressure $V_3\,E$ until the point F is reached. Since the pressure is constant, the temperature is constant, and the line $E\,F$ corresponds to the isothermal compression line of Fig. 232. At F, the exhaust port closes, and the steam is compressed adiabatically during the remainder of the stroke $F\,K$, following the curve $F\,G$. Now, by adding heat, the pressure is raised to $V_1\,B$, and the above operations may be repeated.

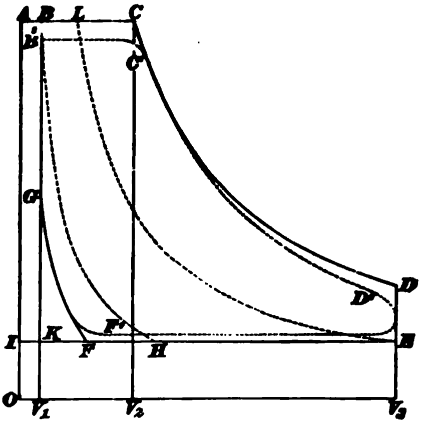


FIG. 234.

It is evident that the cycle just described is not reversible, being open at both ends; but, nevertheless, it has a greater efficiency than could be obtained from an actual engine. The thermal efficiency of the process just described is easily found. The temperature corresponding to a pressure of 100 pounds per square inch is, from the steam table, 327.625°, and to 14.7 pounds per square inch, 212°; whence $T_1 = 787.625°$ and $T_2 = 672°$. Therefore, the efficiency $= \frac{787.625 - 672}{787.625} = 14.68\%$. Since this 14.68% represents the efficiency when the steam operates through a reversible cycle, it is evident that no non-condensing steam engine operating with a boiler pressure of 100 pounds, absolute, can attain an efficiency as high as 14.68%, for perfect conditions can never be obtained, there being no substance which is a perfect non-conductor of heat. The dotted outline $B'\ C'\ D'\ F'\ G$ shows a very good diagram supposed to be taken from an actual engine. Here the initial pressure is $V_1\ B'$, 5 pounds less than the boiler pressure. The back pressure is a little over 2 pounds, say enough to make it 17 pounds, absolute. It will be noticed that all of the corners, except B', are rounded, and that the expansion line $C'\ D'$ falls below the theoretical expansion line $C\ D$. In consequence of this, the engine operates as though the boiler pressure were 95 pounds (corresponding to a temperature of 323.936°) and the back pressure 17 pounds (corresponding to a temperature of 219.452°). Hence, the theoretical thermal efficiency is $\frac{783.936 - 679.452}{783.936} = 13.33\%$.

To show what the conditions must be in order that the steam engine may operate through a reversible cycle, consider Fig. 234 again. It is absolutely necessary that the cycle be closed; hence, the steam must be cut off at some point L so chosen that, during the succeeding adiabatic expansion, the pressure will fall to $V_2\ E$ at the end of the stroke; a point H must be chosen for the point of exhaust closure such that, at the end of the subsequent adiabatic compression, the pressure will be $V_1 B$. In other words, the diagram must be $B L E H B$. With these conditions fulfilled, and with a cylinder which is a perfect non-conductor of heat, the cycle would be reversible, provided there were no rounded corners.

STEAM AND STEAM ENGINES.

STEAM.

1189. **Steam** is water vapor—that is, it is water changed to a gaseous state by the application of heat.

1190. The effects of the application of heat to water may be shown by the following simple experiment: Take a cylinder of indefinite length, fitted with a piston, which, let us suppose, has no weight, and place within the cylinder say 100 cubic inches of water. For convenience, let the area of the cross-section of the cylinder be 100 square inches; then, the height of the water in the cylinder is 1 inch. (See (*a*), Fig. 235.) Assume the temperature of the water to be 32° F.—that is, just at freezing point. The upper end of the cylinder is open to the atmosphere, and the piston and the water under it are, therefore, subjected to a pressure of 14.69 pounds per square inch, or a total pressure of $100 \times 14.69 = 1,469$ pounds. Now, let heat be applied to the water in the cylinder; what will take place?

The temperature will gradually rise from 32°. The volume of the water will decrease slightly, until the temperature reaches 39.2° F., and then will increase gradually, though almost imperceptibly, until the temperature reaches 212° F. At that point the piston will be 1.043 inches from the bottom of the cylinder, instead of 1 inch, the volume of the water thus having increased 4.3%, while the temperature has risen from 32° F. to 212° F. Just at this point the vibrations of the molecules of the water, which have become more and more rapid as the temperature has risen, have become rapid enough not only to overcome the force of cohesion, which tends to keep them together, but also to raise the piston with its load of 1,469 pounds. The water begins to change to the gaseous form, or, in other words, begins to become steam. The temperature now remains at 212°, and can not be raised above that point as long as any water remains in the bottom of the cylinder.

1191. If, after the water is heated to 212°, more heat is added and the pressure remains at 14.69 pounds per square inch, there will be no rise in temperature, but some of the water will change to steam and the piston will rise as shown at (*b*), Fig. 235. Now, push the piston down until the volume of the steam is reduced one-half, withdrawing heat so as to keep the temperature at 212°. We know that, with a perfect gas, this decrease in volume would double the pressure; with steam, however, the only effect is to change part of the steam to water without changing the pressure.

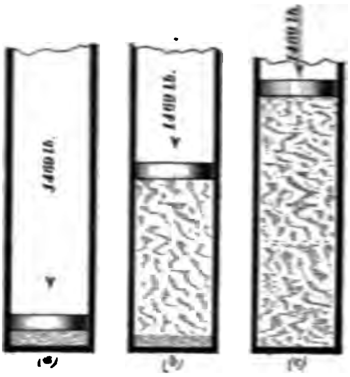


Fig. 235.

Now, raise the piston, keeping the temperature at 212°. With a perfect gas, we know that the pressure would fall in exact proportion to the increase in volume; the steam, however, behaves in a different manner. Part of the water changes to steam as the piston rises, but there is no change in the pressure.

If the pressure on the piston is increased, it is found that the temperature of the water must be increased before steam is formed; thus, make the pressure 30 pounds per square inch, and we find that no steam is formed until the temperature reaches 250.3°. At this temperature, it is also found that no change in the volume of the steam affects its pressure.

We have learned that the pressure of a perfect gas depends on its temperature and volume; the above experiments, however, show that the pressure of steam in contact with water is independent of the volume, but depends entirely on the temperature; if there is no change in temperature, the only effect of a change in volume is to change the quantity of steam.

Steam in contact with water is called **saturated steam.** If not in contact with water, steam is still saturated if its temperature is the same as that at which water boils when subjected to the same pressure.

The conditions of saturation apply as well to all other vapors as to steam; any vapor in contact with the liquid from which it is formed is *saturated*, and for a given temperature, can have but one pressure.

1192. Suppose, now, that just enough heat is applied to change all the water into steam, as in (*c*), Fig. 235. Up to the point where the last drop of water is evaporated, the steam is *saturated*, and at the pressure of 14.69 pounds per square inch, the temperature remains at 212°. After the last drop of water is evaporated, there is a change in the effect produced when heat is applied. Both the temperature and volume of the steam now increase, and in many respects its properties resemble those of a perfect gas. Steam in this condition is said to be **superheated.** The specific heat of superheated steam under constant pressure is .48.

1193. To raise each pound of the water from 32° to 212° required the expenditure of 180.531 B. T. U. If the specific heat of water were 1 at all temperatures, the heat required would have been $212 - 32 = 180$ B. T. U., but at high temperatures the specific heat of water is a little greater than 1, consequently a little more than 180 B. T. U. are required.

The work expended in raising 1 pound of water from 32° to 212° may be found by multiplying the number of heat units required by 778, the mechanical equivalent of heat; it is therefore $180.531 \times 778 = 140{,}453.12$ foot-pounds.

To change each pound of water at 212° to steam at 212°

requires the expenditure of 966.069 B. T. U., or 966.069 × 778 = 751,601.68 foot-pounds of work.

Part of this work was spent in lifting the piston against the atmospheric pressure, and part in overcoming molecular forces. The former is called **outer** or **external work**; the latter, **inner** or **internal work.**

The outer work may be calculated as follows: At the beginning of the operation the piston was 1 inch from the bottom of the cylinder; when all the water had been changed to steam, the piston was 1,646 inches above the bottom. It had therefore been raised 1,646 − 1 = 1,645 inches = $137\frac{1}{12}$ feet, against a constant pressure of 1,469 pounds, and the work done was 1,469 × $137\frac{1}{12}$ = 201,375.417 foot-pounds, of which all but the very small portion represented by the expansion of the water when heated from 32° to 212° was developed during the formation of the steam. At a temperature of 32°, 100 cubic inches of water weigh 3.6121 pounds; hence, the external work per pound of water was 201,375.417 ÷ 3.6121 = 55,750.2 foot-pounds. The total work of changing 1 pound of water to steam at 212° was 751,601.68 foot-pounds; of this, nearly 55,750.2 foot-pounds were spent in external work, leaving 751,601.68 − 55,750.2 = 695,851.48 foot-pounds as the internal work.

1194. Let the experiment be tried again with tne same amount of water in the cylinder, but with a weight on the piston in addition to the atmospheric pressure. Let the pressure be say 32 pounds per square inch, the total pressure being 32 × 100 = 3,200 pounds. It will now be found that steam does not begin to form until the temperature has reached 254° F. This is easily accounted for, since the vibrations of the molecules of the water would necessarily have to be more rapid, that is, the temperature would have to be higher, in order to overcome the increased pressure on the piston.

It will now require 223.021 B. T. U. = 173,510.34 foot-pounds to raise 1 pound of the water from 32° to 254°. It will require 936.389 B. T. U. = 728.510,64 foot-pounds to change one pound of the water at 254° into steam of the same temperature. When the water is all evaporated, the piston will be found at a height of 791.8 inches from the bottom of the cylinder. It has been raised by the steam to a height of $791.8 - 1 = 790.8$ inches $= 65.9$ feet, against a constant pressure of 3,200 pounds. The total outer work is, therefore, $65.9 \times 3{,}200 = 210{,}880$ foot-pounds. The outer work per pound of water is $\frac{210{,}880}{3.6121} = 58{,}381.55$ foot-pounds. The inner work is $728{,}510.64 - 58{,}381.55 = 670{,}129.09$ foot-pounds.

1195. It is thus apparent that changing the pressure on the piston has changed all the other conditions—the temperature, the volume, the inner and outer works, and the heat required to raise the water from 32° to the boiling point.

The relations between these various properties of steam can not be expressed by any general law. They have been determined experimentally, however, and, taken together, form what are known as the steam tables.

STEAM TABLES.

RELATION BETWEEN PRESSURE AND TEMPERATURE OF SATURATED STEAM.

1196. It has been shown that the temperature of saturated steam depends *only* upon the pressure, and not at all upon the volume or any other property of the steam; conversely, the pressure depends upon the temperature.

If the steam in a boiler shows a certain pressure by the gauge, we know that it can only have one temperature. In fact, it would be possible, though, perhaps, not convenient, to determine the pressure of steam from the reading of a thermometer inserted into it. At the atmospheric pressure of 14.69 pounds per square inch, saturated steam *always* has a temperature of 212° F.; at a pressure of 79 pounds per square inch above vacuum, steam can have only a temperature of 311° F.

1197. The temperatures of saturated steam corresponding to the different pressures have been experimentally determined with great care and accuracy by the French physicist, Regnault, and are given in the Table of the Properties of Saturated Steam. Column 1 of the steam table contains the pressures from 1 to 300 pounds per square inch, and column 2 contains the corresponding temperatures.

A large number of formulas have been devised to express the relation between the pressure and temperature of saturated steam. The following by Rankine is probably as accurate and convenient as any:

Let t = temperature, Fahrenheit;
T = absolute temperature $= t + 460°$;
p = absolute pressure (pressure above vacuum) in pounds per square inch.

$$\text{Then, } \log p = 6.1007 - \frac{2{,}719.78}{T} - \frac{400{,}215}{T^2}. \qquad \textbf{(90.)}$$

Example.—Find the pressure corresponding to 285.8° *F.*

Solution.—$T = 285.8 + 460 = 745.8°$.

$$\text{Log } p = 6.1007 - \frac{2{,}719.78}{745.8} - \frac{400{,}215}{745.8^2} = 1.73437.$$

The number corresponding to this logarithm is 54.246. Hence, the pressure is 54.246 lb. per sq. in. The table gives 54 lb.

EXAMPLES FOR PRACTICE.

1. The temperature of steam being 224°, what is the pressure?
Ans. 18.557 lb. per sq. in.

2. Find the pressure in the last example by using formula **90**.
Ans. 18.574 lb. per sq. in.

3. Find the pressure corresponding to a temperature of 312°, (*a*) by means of the steam tables; (*b*) by formula **90**.
Ans. (*a*) 80.157 lb. per sq. in.
(*b*) 80.566 lb. per sq. in.

4. The pressure being 105 pounds absolute, (*a*) find the temperature by formula **90**; (*b*) what is the difference between the above result and the value given in the tables?
Ans. (*a*) 330.756°.
(*b*) .418°.

5. Assuming that there has been no heat added or taken away, what is the difference in pressure due to a fall of temperature from 281.7° to 259°? Ans. 15.764 lb. per sq. in.

NOTE.—All examples under the heading *Examples for Practice* should be worked out by the student. The student should solve them in all cases before proceeding further.

1198. Heat of the Liquid.—Column 3 of the table gives the heat required to raise *one pound* of water from 32° F. to the temperature given in column 2. It will be observed that it is nearly the same as the difference between the given temperature and 32°. For instance, the heat of the liquid corresponding to a pressure of 60 pounds, and a temperature of 292.575°, is 262.248 B. T. U. The difference in temperatures is $292.575° - 32° = 260.575°$. In many steam tables this column is left out, and the heat of the liquid is found for any temperature by subtracting 32° from the temperature. This method is usually exact enough for practical purposes. The heat required to raise the temperature of a pound of water from 32° to the boiling point is called the **heat of the liquid.**

The results of column 3 have been determined experimentally.

1199. The **total heat of vaporization** for any given temperature is the number of heat units required to raise a pound of water from 32° F. to the given temperature, and change it into steam at that temperature. In the first experiment, shown in Fig. 235, the total heat of vaporization was the total heat applied to the cylinder from the beginning of the experiment until the water was all turned into steam. It is the sum of the heat of the liquid, the internal work, and the external work.

Referring back to the first experiment, the heat of the liquid was found to be 180.531 B. T. U. per pound, or, expressed in units of work, 140,453.12 foot-pounds. The heat used to perform outer work was found to be equivalent to 55,750.2 foot-pounds, and that used in performing inner work was equivalent to 695,851.48 foot-pounds. The total heat of vaporization, then, is equivalent to the sum of all these works, or to 140,453.12 + 55,750.02 + 695,851.48 = 892,054.8 foot-pounds, or, expressed in thermal units, $\frac{892,054.8}{778} = 1,146.6$ B. T. U., the same as given in the steam table.

In the second experiment, the work equivalent to the heat of the liquid was 173,510.34 foot-pounds, the outer work 58,381.55 foot-pounds, and the inner work 670,129.09 foot-pounds. The total heat of vaporization (expressed in foot-pounds) is, therefore,

$58,381.55 + 173,510.34 + 670,129.09 = 902,020.98$ foot-pounds,

or $\frac{902,020.98}{778} = 1,159.41$ B. T. U. per pound.

1200. The total heat of vaporization has also been experimentally determined by Regnault for all pressures and temperatures. The following formula expresses the results of the experiments very closely:

$$H = 1,081.94 + .305\ t, \qquad \textbf{(91.)}$$

in which H is the total heat of vaporization, expressed in heat units, and t the temperature in degrees F.

EXAMPLE.—What is the total heat of a pound of steam when the absolute pressure is 50 pounds per square inch?

SOLUTION.—Looking in column 2, the temperature corresponding to a pressure of 50 pounds is 280.904° F. Substituting this in formula **91**,

$H = 1,081.94 + .305 \times 280.904° = 1,167.615$ B. T. U. Ans.

The **total heat of vaporization** is given in column 5 of the steam table.

1201. Latent Heat of Vaporization.—In the first experiment it was found that a certain number of B. T. U.'s were required to raise a pound of water from 32° to the temperature of the boiling point, and that another quantity of heat was required to change the water at boiling point into steam at the same temperature. The former of these quantities of heat we called the heat of the liquid; the latter is the **latent heat of vaporization**, or simply the **latent heat of steam.** The heat required for both of the above operations was defined as the **total heat.** Therefore, letting H represent the total heat, q the heat of the liquid, and r the latent heat of vaporization, we have the following equation:

$$H = q + r.$$

$$\text{Therefore, } r = H - q.$$

Hence, to find the latent heat of vaporization, subtract the heat of the liquid from the total heat. The latent heat is given in column 4 of the table, and is obtained by simply subtracting the numbers in column 3 from those in column 5.

EXAMPLES FOR PRACTICE.

1. What is the total heat of 25 pounds of saturated steam having a temperature of 340°? Ans. 29,640.975 B. T. U.

2. What is the total heat of a pound of steam at a pressure of 90 pounds gauge? Ans. 1,182.88 B. T. U.

3. Solve both of the above problems by means of formula **91.**

Ans. { 29,641 B. T. U.
1,182.882 B. T. U.

4. What is the latent heat of vaporization corresponding to a pressure of 85 pounds absolute? Ans. 892.086 B. T. U.

5. Find the heat of the liquid of 10 pounds of water at a temperature of 297.6°. Ans. 2,673.77 B. T. U.

6. What is the latent heat of vaporization of the water in example 5? Ans. 9,053.31 B. T. U.

1202. The **specific volume** of saturated steam has been determined both experimentally and by complicated mathematical formulas. Column 7 of the table gives the volume in cubic feet of a pound of steam at the different pressures, and column 8 the ratio between the volume of a pound of steam at the temperature in column 2 and a pound of water at maximum density.

1203. The results in column 7 may be found approximately by Rankine's formula:

$$p\,V^{\frac{17}{16}} = 475, \qquad \textbf{(92.)}$$

in which p is the pressure in pounds per square inch, and V is the volume in cubic feet of a pound of steam at the given pressure.

EXAMPLE.—Find the volume of one pound of steam at a pressure of 30 pounds per square inch absolute.

SOLUTION.—

$$30 V^{\frac{17}{16}} = 475$$

$$V^{\frac{17}{16}} = \frac{475}{30} = 15.83$$

$$\log V^{\frac{17}{16}} = \frac{17}{16} \log V = \log 15.83 = [illegible]$$

$$\log V = [illegible] \div \frac{17}{16} = [illegible]$$

$V =$ [illegible] cubic feet. The table gives [illegible] cubic feet.

Column 8 may be obtained from column 7 by multiplying by 62.425, the weight in pounds of a cubic foot of water.

EXAMPLES FOR PRACTICE.

1. Find, by means of the tables, the volume of 12 pounds of saturated steam having a gauge pressure of 100 pounds. Ans. 45.42 cu. ft.

2. Solve the above example by formula **92.** Ans. 45.336 cu. ft.

3. The volume of 5 pounds of saturated steam is 22 cubic feet: (*a*) find the pressure by formula **92**; (*b*) find the weight corresponding to this pressure from the tables.
Ans. (*a*) 98.405 lb. per sq. in.
(*b*) .226829 lb. per cu. ft.

4. If 2.87 cubic feet of steam expand until the volume and pressure are, respectively, 9 cubic feet and 2½ pounds gauge, what was the original pressure? Ans. 57.93 lb. per sq. in.

5. The steam in an engine cylinder has a pressure of 85 pounds gauge, and a volume of 1.6 cubic feet at cut-off. It expands to a pressure of 18 pounds absolute; what is the final volume? Use formula **92,** in both this and the last example, remembering that if $p V^{\frac{17}{16}} = 475$, $p_1 V_1^{\frac{17}{16}} = 475$ also, and, consequently, that $p V^{\frac{17}{16}} = p_1 V_1^{\frac{17}{16}}$.
Ans. 8.0134 cu. ft.

1204. The **density of saturated steam** (that is, the weight in pounds of a cubic foot for the different pressures and temperatures) is given in column 6.

The density is the reciprocal of the specific volume. For example, a pound of steam at 44 pounds pressure occupies a volume of 9.403 cu. ft.; therefore, one cubic foot of steam at 44 pounds pressure must weigh $\frac{1}{9.403} = .1063$ pound. Hence, the values in column 6 may be obtained by taking the reciprocals of those in column 7.

1205. The above properties of saturated steam compose the Table of the Properties of Saturated Steam. Two other properties are given in some steam tables; viz., the heat corresponding to the inner work, and the heat corresponding to the outer work, the sum of the two being the latent heat of vaporization. As these properties are not of much practical importance, they have been omitted from the table.

It must always be borne in mind that, in using this table, the pressures are reckoned from vacuum, and *not* from atmospheric pressure. That is, 14.7 pounds must be added to all gauge pressures to make them available for use in the steam table. Again, the heat of the liquid, the total heat, and the latent heat are calculated from 32° F., *not* 0° F. A great deal of trouble will be saved by carefully remembering these points.

EXAMPLES ON THE USE OF THE STEAM TABLE.

1206. Example 1.—Find the heat required to change 6 pounds of water at 32° into steam at 84 pounds pressure absolute.

Solution.—The total heat of one pound at 84 pounds pressure is, from column 5, 1,178.091 B. T. U.

1,178.091 × 6 = 7,068.546 B. T. U. Ans.

Example 2.—How many heat units are required to raise 8½ pounds of water from 32° to 250° F.?

Solution.—Looking in column 3, the heat of the liquid of one pound at 250.293° is 219.261 B. T. U. 219.261 − .293 = 218.968 B. T. U. $= q$ for 250°.

Then, for 8½ pounds, it is 218.968 × 8½ = 1,861.228 B. T. U. Ans.

Example 3.—How many foot-pounds of work will it require to change 60 pounds of water at 311° F. into steam at the same temperature?

Solution.—Looking under column 4, the latent heat of vaporization is 895.729; that is, it takes 895.729 B. T. U. to change one pound of water at 311° into steam at the same temperature. Therefore, it takes 895.729 × 60 = 53,743.74 B. T. U. to perform the same operation upon 60 pounds of water.

53,743.74 B. T. U. = 41,812,629.73 foot-pounds. Ans.

Example 4.—Find the volume occupied by 14 pounds of steam at 30 pounds *gauge* pressure.

Solution.—30 pounds gauge pressure = 30 + 14.7 = 44.7 pounds absolute pressure. The nearest pressure in the table is 44 pounds, and the volume of a pound at that pressure is 9.403 cu. ft. The volume of a pound at 46 pounds pressure is 9.018 cu. ft. 9.403 − 9.018 = .385 cu. ft., the difference in volume for a difference in pressure of 2 lb. $\frac{.385}{2} = .1925$ cu. ft., the difference in volume for a difference in pressure of 1 pound. .1925 × .7 = .135 cu. ft., the difference in volume for a difference in pressure of .7 pound. Therefore, 9.403 − .135 = 9.268 cu. ft. is the volume of one pound of steam at 44.7 pounds pressure.

9.268 × 14 = 129.752 cu. ft. Ans.

Example 5.—Find the weight of 40 cubic feet of steam at a temperature of 254° F.

Solution.—The weight of one cubic foot of steam at 254.002°, from the table, is .078839 pound. Neglecting the .002°, the weight of 40 cu. ft. is, therefore, .078839 × 40 = 3.15356 pounds. Ans.

Example 6.—How many pounds of steam at 64 pounds pressure absolute are required to raise the temperature of 300 pounds of water from 40° to 130° F., the water and steam being mixed together?

Solution.—The number of heat units required to raise one pound from 40° to 130° is 130° − 40° = 90 B. T. U. (Actually a little more than 90 would be required, but the above is near enough for all practical purposes.) Then, to raise 300 pounds from 40° to 130° requires 90 × 300 = 27,000 B. T. U. This quantity of heat must necessarily come from the steam. Now, one pound of steam at 64 pounds pressure gives up, in condensing, its latent heat of vaporization, or 905.9 B. T. U.; but, in addition to its latent heat, each pound of steam on condensing must give up an additional amount of heat in falling to 130°. Since the original temperature of the steam was 296.805° F. (see table), each pound gives up by its fall of temperature 296.805 − 130 = 166.805 B. T. U. Consequently, each pound of the steam gives up a total of

905.9 + 166.805 = 1,072.705 B. T. U.

It will, therefore, take $\frac{27{,}000}{1{,}072.705} = 25.17$ pounds of steam to accomplish the desired result.

EXAMPLES FOR PRACTICE.

1. How many foot-pounds of work are required to change 42 pounds of water at 319° into steam at a temperature of 330°?
Ans. 29,190,679.8 ft.-lb.

2. How many B. T. U. are required to convert 25 pounds of water at 32° into 109.6 cubic feet of steam? Ans. 29,541.15 B. T. U.

3. Find the number of heat units required to change 11 pounds of water at 50° into steam at 100 pounds absolute pressure.
Ans. 12,802.526 B. T. U.

4. Find the weight of 712 cubic feet of steam at a pressure of 33 pounds gauge. Ans. 81.712 lb.

5. How many cubic feet of steam at 47.3 pounds pressure, gauge, are required to raise 120 pounds of water from 55° to 160° at atmospheric pressure? Ans. 82.359 cu. ft.

6. Find the volume of 19 pounds of steam at a temperature of 274°.
Ans. 175.883 cu. ft.

1207. The crude apparatus used in the first experiment above described (the cylinder containing the water and the piston, surrounded by fire) constitutes the simplest form of a steam engine. By means of it, heat may be transformed into work. It is certainly a very poor and inconvenient form of engine, since the only work it can do is that of raising weights; and, again, the fire must be successively removed and renewed every time a weight is lifted.

It will be seen that this engine consists of four essential elements; viz.: the fire, or source of heat; the water, or medium for the transfer of heat; the cylinder, with a piston to prevent the steam from escaping into the air, and a source of cold. Nearly all engines in actual operation consist of the above four elements, more advantageously arranged. In the actual engine, the fire is not in contact with the cylinder, but the steam is generated in a separate vessel known as the **boiler.** Again, the cylinder is of a fixed length, and the steam usually acts on both sides of the piston, instead of on one side.

This improvement of the actual engine over the simple apparatus of the first experiment leads to the consideration of some further essential characteristics of steam.

diagram represent a pressure of 30 pounds. Then, the length $O\,A$, which represents the steam pressure in question, must have a length of $\frac{60}{30} = 2$ inches; and, since the pressure is constant for all positions of the piston, the lengths *1-1'*, *2-2'*, *3-3'*, *4-4'*, etc., must all be equal to 2 inches. In other words, the line of pressures $A\,B$ must be parallel to the line $O\,X$. It was observed above that the temperature of the steam is also constant throughout the stroke of the piston. Hence, the above line $A\,B$ is the *isothermal* (constant temperature line) of saturated steam. Hence, it follows that *the isothermal of saturated steam is a straight line parallel to the axis of volumes O X.*

1211. The pressure $O\,A$ on the left of the piston is taken from vacuum. There is also a pressure on the right of the piston of 14.7 pounds per square inch, since the right end of the cylinder is open to the atmosphere. Let this be represented by the height $O\,D = \frac{14.7}{30} = .49$ in. Since this atmospheric pressure is constant throughout the stroke of the piston, it may be represented by the straight line $D\,C$ parallel to $O\,X$. The *net pressure* on the piston is then represented by

$$O\,A - O\,D = D\,A = 60 - 14.7 = 45.3 \text{ lb. per sq. in.}$$

The work done by the steam during one stroke of the piston may now be easily found.

There is a constant net pressure throughout the stroke of 45.3 lb. per sq. in. $= 45.3 \times 144 = 6{,}523.2$ lb. per sq. ft. The total pressure on the piston is, therefore, $6{,}523.2 \times 2 = 13{,}046.4$ lb. Work is defined as pressure, force, or resistance, multiplied by the distance through which it acts. Consequently, the work spent in moving the piston from one end of the cylinder to the other, a distance of 6 feet, is $13{,}046.4 \times 6 = 78{,}278.4$ foot-pounds.

1212. For our purpose, the work may be obtained in a more convenient way. As shown above, the pressure on the piston per square foot is 6,523.2 pounds. The volume of the cylinder = area × length = 2 sq. ft. × 6 ft. = 12 cu. ft. Now, multiplying the pressure in pounds per square foot by the volume of the cylinder in cubic feet, the result is 6,523.2 × 12 = 78,278.4 foot-pounds, as before.

Let P = the pressure on the piston in lb. per sq. ft.;
p = pressure on piston in lb. per sq. in.;
V = volume in cubic feet swept through by the piston;
W = work in foot-pounds.

Then, $$W = P\,V = 144\,p\,V. \qquad \textbf{(93.)}$$

Referring to Fig. 237, it will be remembered that the line $D\,A$ represents the pressure p, and the line $D\,C$ represents the volume V. The product $D\,A \times D\,C =$ area of the rectangle $A\ B\ C\ D$. But $A\ D \times D\ C = p\,V$. *Therefore, the area of the rectangle is proportional to the work of the steam.*

$D\,A = \frac{45.3}{30} = 1.51$ inches. $D\,C = 2$ inches. Then, the area $A\ B\ C\ D = 1.51 \times 2 = 3.02$ sq. in. But 1 inch in height = 30 lb. per sq. in., and one inch in length $\frac{12}{2} = 6$ cu. ft. Hence, $p\,V = 3.02 \times 30 \times 6 = 543.6$ foot-pounds, and $W = 144\,p\,V = 543.6 \times 144 = 78{,}278.4$ foot-pounds.

1213. In the same way it can be shown that, no matter what the form of the area $A\,B\,C\,D$ may be, it will represent the work done by the steam, provided the upper line $A\,B$ represents the relation between the pressures and volumes of the steam on one side of the piston, and the lower line $C\,D$ represents the relation between the pressures and volumes on the other side. For, as shown in Art. **1158**, the area is equal to the length of the diagram multiplied by the mean ordinate, and the mean ordinate simply represents the average net pressure on one side of the piston.

Hence, in general, to find the work of the steam from the diagram, *multiply the area in square inches by the vertical scale of pressures, by the horizontal scale of volumes, and by 144. The result is the work in foot-pounds.*

EXAMPLE.—The area of a diagram of the character shown in Fig. 237 is 7.34 sq. in. The vertical scale of pressures is 36 pounds to the inch, and the horizontal scale of volumes is 2½ cu. ft. to the inch. What is the work of the steam per stroke of piston?

SOLUTION.—Work = 7.34 × 36 × 2½ × 144 = 95,126.4 ft.-lb.

1214. Suppose now that the left end of the cylinder is shut off from the boiler just as the piston in Fig. 237 has completed $\frac{1}{3}$ of the stroke—that is, when it is just 2 feet from the left end of the cylinder. The pressure on the left of the piston, as in Fig. 237, will be constant so long as the cylinder is in communication with the boiler—that is during the first 2 feet of the stroke. Therefore, the pressure line for the first $\frac{1}{3}$ of the diagram will be a straight line *A B* parallel to *O X* (see Fig. 238), and, as before, $\frac{60}{30} = 2$ inches above it. At this point the steam supply is shut off, and the 4 cubic feet of steam in the cylinder must expand as the piston moves along, until it fills the entire volume of the cylinder. Its pressure, then, must fall somewhat after the manner of a perfect gas. This fall in pressure is graphically represented by the curved line *B E*. Assuming for the moment that the curve is the ***equilateral hyperbola*** (see Art.

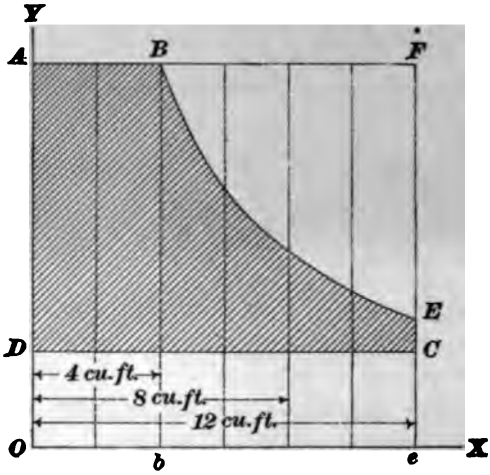


FIG. 238.

1161), the work may be calculated as follows: The height of Fig. 238 is 2 inches; the length $A\ B$ is $\frac{2}{3}$ inch. Therefore, the area $A\ B\ b\ O = 2 \times \frac{2}{3} = 1.3333$ sq. in.

1215. The area $B\ E\ e\ b$ may be found by the following formula, in which

A = area of diagram in square inches;
p_1 = initial pressure (as $O\ A$) measured in inches;
V_1 = volume at cut-off, measured in inches.

$$A = 2.3026 \times p_1\, V_1 \log \frac{V_2}{V_1} \qquad \textbf{(94.)}$$

$$= 2.3026 \times \frac{2}{3} \times 2 \times \log \frac{6}{2} = 1.4648 \text{ sq. in.}$$

Then, area $A\ B\ E\ e\ O = 1.3333 + 1.4648 = 2.7981$ sq. in.
$O\ D = .49$ in. Area $C\ D\ O\ e = .49 \times 2 = .98$ sq. in.
Area $A\ B\ E\ C\ D = A\ B\ E\ e\ O - C\ D\ O\ e = 2.7981 - .98 = 1.8181$ sq. in.

Then, $L = 144\, a\, h\, h_1$, **(95.)**

in which, L = work in foot-pounds;
a = net area of diagram (as $A\ B\ E\ C\ D$);
h = scale used to lay off pressures;
h_1 = scale used to lay off volumes, the volumes being always expressed in cubic feet, and the pressures in pounds per square inch.

In the above diagram

$L = 1.8181 \times 30 \times 6 \times 144 = 47{,}125.15$ ft.-lb.

The work done in the first case was 78,278.4 ft.-lb.; so by shutting off the steam at $\frac{1}{3}$ of the stroke the work of the steam has been decreased by $78{,}278.4 - 47{,}125.15 = 31{,}153.25$ ft.-lb. But, in the first case, 12 cu. ft. of steam at boiler pressure were used, while, in the last case, only 4 cu. ft. were used; hence, the work per cubic foot of steam was, in the first case, $\frac{78{,}278.4}{12} = 6{,}523.2$ ft.-lb., and, in the second case, $\frac{47{,}125.15}{4} = 11{,}781.29$ ft.-lb., or nearly twice as much. This shows the economy of cutting off the steam early in the stroke and allowing it to expand.

EXAMPLES FOR PRACTICE.

1. The mean ordinate of a diagram, similar to that shown in Fig. 238, is 1.2 inches long. The vertical scale of pressures is 1" = 40 lb. per sq. in., and the horizontal scale of distances is 1" = 10". The length of the diagram is 3", and 1 ft. of actual length of the vessel which contains the steam represents a volume of 452 cu. in. What is the work done in one stroke of the piston? Ans. 4,520 ft.-lb.

2. The mean pressure in the cylinder of a steam engine is 38.7 lb. per sq. in.; the diameter of the cylinder, 26"; the stroke, 32", and number of strokes per minute, 120. Find the horsepower by means of formula **93.** Ans. 199.244 H. P.

3. The mean ordinate of a diagram is .89"; the length, 3.2"; the vertical scale of pressure, 1" = 50 lb. per sq. in.; the horizontal scale of volumes, 1" (diagram) = .56 cu. ft. Find the work done in 12 strokes. Ans. 137,797.6 ft.-lb.

1216. The Expansion Line.—The character of the expansion curve *B E*, Fig. 238, depends upon a variety of circumstances. Suppose, first, that the steam expands without any condensation. The equation representing the relation between the pressures and volumes will then be

$$p\,V^{\frac{17}{16}} = \text{some constant.}$$

When the quantity of the expanding steam is one pound, the constant is 475, and the equation becomes

$$p\,V^{\frac{17}{16}} = 475,$$

which has already been given. The curve may be plotted by laying off the volumes, taken from column 7 of the steam table, along the line *O X*, and the corresponding pressures from column 1, along the line *O Y*. It will now be necessary to determine the conditions under which the steam will follow the above expansion line.

1217. Suppose that one pound of steam at 60 pounds pressure (absolute) be confined in a non-conducting cylinder. Its volume, according to the table, is 7.024 cubic feet. Let the pound of steam expand until the pressure reaches 54 pounds absolute; the volume, then, is 7.756. It may be assumed with little error that the portion *a b*, Fig. 239, of the expansion line, which represents the drop of pressure from 60 to 54 pounds, is straight. The average pressure is, then, $\frac{60 + 54}{2} = 57$ pounds.

The volume dc is $7.756 - 7.024 = .732$ cu. ft.

The work $= 144\,p\,V = 144 \times 57 \times .732 = 6{,}008.256$ ft.-lb.

The total heat of the steam at 60 pounds pressure is 1,171.176 B. T. U. per pound, and at 54 pounds pressure, 1,169.102 B. T. U. per pound. Hence, the pound of steam during its expansion gives up $1{,}171.176 - 1{,}169.102 = 2.074$ B. T. U. $= 1{,}613.572$ foot-pounds.

1218. The work done by the expanding steam is 6,008.256 foot-pounds, while the work given up by the steam during its expansion is only 1,613.572 foot-pounds, or a little over $\frac{1}{4}$ as much. It is evident that the work could not have been done by the expansion of the steam alone. There remains $6{,}008.256 - 1{,}613.572 = 4{,}394.684$ foot-pounds of work that must have been obtained in some other manner. The cylinder is non-conducting, so no heat could have been obtained from without. The *only* way in which this extra amount of work can be accounted for is to suppose that a small amount of steam is condensed while expanding, and thus gave up its latent heat of vaporization. 4,394.684 foot-pounds $= 5.649$ B. T. U. The average pressure was 57 pounds, and the latent heat of steam at that pressure is

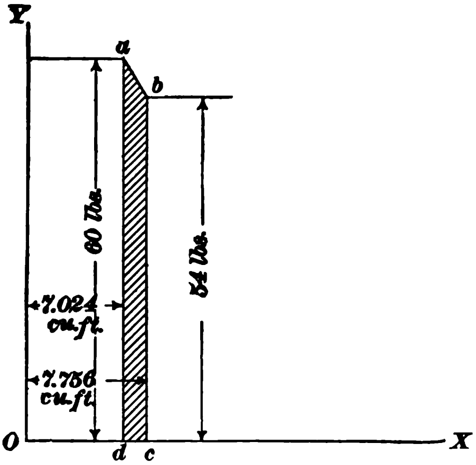


FIG. 239.

911.304 B. T. U. Hence, $\frac{5.649}{911.304} = .0062$ of the pound of steam must have condensed to furnish the extra 4,394.684 foot-pounds of work. We thus arrive at the conclusion that *saturated steam expanding in a non-conducting cylinder must of necessity condense somewhat.*

In the same manner it can be shown that, if saturated steam be compressed in a non-conducting cylinder, it must become superheated.

1219. It is thus apparent that steam in a non-conducting cylinder cannot expand according to Rankine's formula $pV^{\frac{17}{16}} =$ a constant. The curve is really the adiabatic for steam, and is usually represented by the formula,

$$pV^{\frac{10}{9}} = \text{a constant.}$$

The cylinder of the actual engine is never perfectly non-conducting. In fact, it is made of cast iron, which is a very good conductor of heat. This fact leads to great complication in the study of the action of the steam in the cylinder.

1220. Suppose, first, that the cylinder be simply exposed to the atmosphere—that is, it is not covered with any non-conducting material. The steam, entering one end of the cylinder with a pressure of say 80 pounds absolute, has a temperature of about 312°. As it meets the cold cylinder walls, it will raise them to 312°, nearly; but, at the same time, the steam, meeting a surface cooler than itself, must partially condense. At some point near the middle of the stroke, the communication with the boiler will be cut off, and the steam will expand, the pressure lowering to say 20 pounds absolute, and the temperature, consequently, lowering to about 228°. Therefore, at the end of the stroke, the cylinder walls will be at a temperature of about 228°. The fresh steam now enters the other end of the cylinder with a pressure of 80 pounds and a temperature of 312°, comes in contact with the cylinder walls, and partially condenses, this action being repeated at every stroke. In addition to this condensation, due to the alternate heating and cooling of the cylinder walls, there is the condensation which must always accompany expansion, even in a non-conducting cylinder. Of course, this condensation is almost a total loss, since the water can exert no pressure on the piston.

Some of the condensed steam, however, may be, and generally will be, evaporated again towards the end of the stroke. Just before the steam supply is cut off, the temperature of the cylinder walls is 312°; as the steam expands, it partially condenses and its temperature falls. The temperature of the cylinder walls also falls, and in doing so, gives up heat, part of which is radiated and lost, and part re-evaporates some of the condensed steam. As a consequence of this action, the real expansion line falls below the theoretical line ($p\ V^{\frac{10}{9}} = constant$) just after cut-off, but rises above it towards the end of the stroke.

1221. A very common way of avoiding the loss due to the condensation of steam in the cylinder is to surround the latter with a **steam jacket**—that is, to keep the cylinder continually immersed in steam at boiler pressure. Heat is then transferred from the steam outside the cylinder to the expanding steam inside, and condensation is thus prevented, or, rather, it takes place in the jacket instead of in the cylinder. The expansion curve in this case approaches more nearly the theoretical curve, but usually rises a little above it towards the end of the stroke.

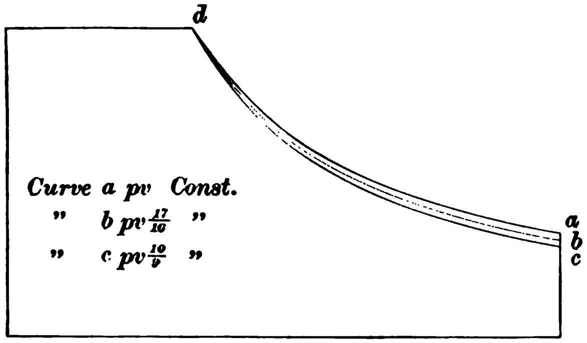


FIG. 240.

1222. Fig. 240 shows the three curves above considered. The upper one is the **equilateral hyperbola,** $p\ V = a$

constant; the second is the **curve of constant steam weight,** $p\,V^{\frac{17}{16}} = a\ constant$, and the third is the **adiabatic** for steam, $p\,V^{\frac{10}{9}} = a\ constant$.

It will be seen how marked may be the difference between the expansion curves under varying circumstances. It is usual, in practice, to assume that the expansion curve is the equilateral hyperbola. And, though it is in no sense the theoretical curve, it is generally nearer than the others, and rather easier to deal with. Hence, in all farther discussions, it will be assumed that expanding steam follows the equilateral hyperbola.

THE STEAM-ENGINE MECHANISM.

THE FOUR-LINK SLIDER CRANK.

1223. It has been shown how steam may do work by lifting weights against the pressure of the atmosphere; since, however, it is not generally desired to do work in this manner, it is essential that some method of changing the to-and-fro motion of the piston into a continuous motion in one direction should be devised. The form of mechanism used for this purpose in practically all types of engines is shown in Fig. 241. It is technically known as the **four-link slider crank.** The steam from the boiler enters one end—say, in this case, the end *h*—of the cylinder, and pushes the piston to the other end. By means of another mechanism, called the **valve,** the steam is now admitted to the end *c* of the cylinder, while the end *h* is at the same time allowed to communicate with the atmosphere or with a condenser.

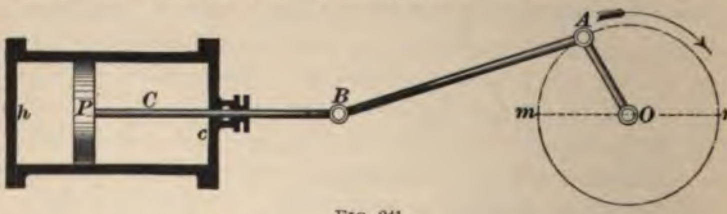


FIG. 241.

The steam in *h* escapes, while that in *c* pushes the piston back again to its original position, whence the same operation is repeated.

Attached to the piston, and forming a part of it, is the piston rod *C B*; to the end of *C B* is fastened, by a joint, one end of the link *B A*. The other end of *B A* is joined to the link *A O*; and the other end of *A O* terminates in a shaft *O*, which is fixed in stationary bearings. It is evident that the end of *B A*, which is attached to *C B*, can move only in a straight line; and since the shaft *O* can rotate only in its bearings, the end of *A O*, which is attached to *B A*, can move only in a circle.

When the piston is at one extreme end of the cylinder, say at *h*, the joint *A* is at the point *m*, and all three links, *A O*, *B A*, and *C B*, lie in a straight line. As the piston moves to the right, the link *B C* moves also to the right, while the joint *A* is constrained to move in the upper semicircle *m n*; when *P* arrives at the other end of the cylinder, the joint *A* is at *n*, and again *A O*, *B A*, and *C B*, are in a straight line. The piston now moves back to the end *h* of the cylinder, the joint *A* moving in the lower semicircle from *n* to *m*.

1224. The link *A O* is called the **crank,** *B A* the **connecting-rod,** and *C B* the **piston rod.** Those parts which have a to-and-fro or reciprocating motion are called the **reciprocating parts.**

The end *h* of the cylinder is called the **head end,** and *c* the **crank end.** The distance passed through by the piston during a half revolution of the crank is called the **stroke;** the stroke is evidently equal to the diameter *m n* of the circle described by the end *a* of the crank.

The engine may run in the direction shown by the arrows in the figure, or it may run in the reverse direction. In the former case it is said to **run over,** and in the latter case to **run under.**

The stroke from the head to the crank end of the cylinder—that is, from left to right in the figure—is called the **for-**

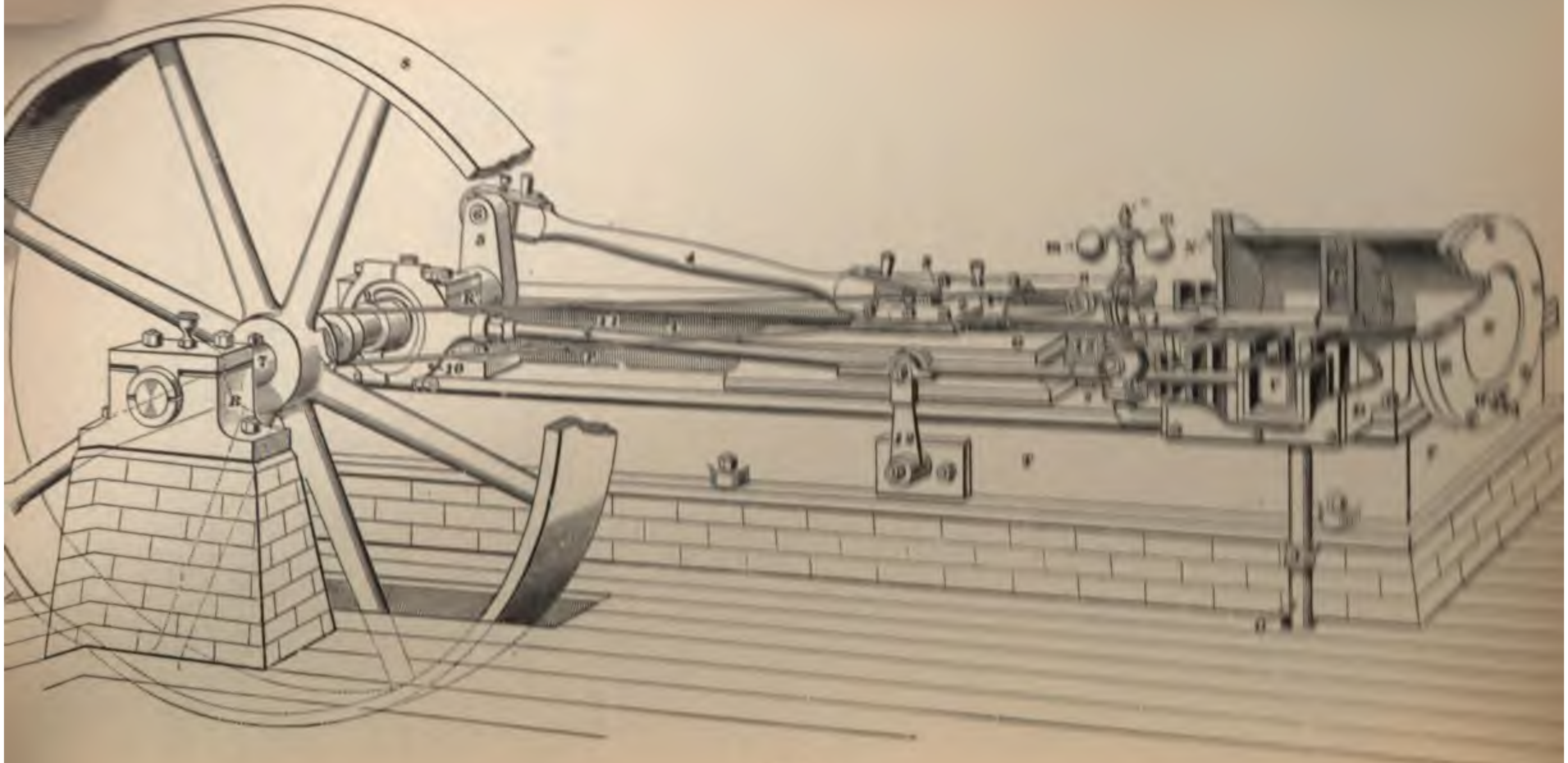


FIG. 242.

ward stroke; the one from crank end to head end, the **return stroke.**

The above simple mechanism perfectly fulfils the office of giving a continuous motion in one direction. A pulley is keyed to the shaft *O*, and the power is transferred by belting from the pulley to shafting, or directly to the machinery to be run.

THE PLAIN SLIDE-VALVE ENGINE.

1225. The enormous number of types of the steam engine may be classified

(1) According to the kind of service, as	Stationary, Locomotive, Marine, etc.
(2) According to number and arrangement of cylinders, as	Simple, Compound, Triple expansion, Quadruple expansion, Duplex, etc.
(3) According to the type of valve used, as	Plain slide-valve, Automatic cut-off, Corliss, etc.

They may all be *horizontal* or *vertical*, *condensing* or *non-condensing*, *single-acting* or *double-acting*. All of these different types involve essentially the same principles, and, therefore, the description of a single type will be sufficient to give a general knowledge of these principles. For this purpose we shall choose the **simple slide-valve engine.**

1226. In Fig. 242 such an engine is shown, and in Fig. 243 is shown a section of a steam cylinder. Referring to these figures, *H* is the head end, and *C* the crank end of the steam cylinder; *B* and *B'* are the steam ports; *D* is the steam chest; *E* is the exhaust port; *N* and *N'* are the cylinder heads; *S* is the steam supply pipe; *O* is the exhaust pipe, and connects with the exhaust port *E*; *G* is one of the two guide bars; *R* and *R'* are the shaft bearings, and *T* is the bed, or frame, of the engine. The above are all stationary parts of the engine, or parts which do not change their relative positions when the engine is in motion. *P* is the piston; *1* is the piston rod; *2* is the cross-head; *3* is the cross-head pin; *4* is the connecting-rod; *5* is the crank; *6* is the crank-pin; *7* is the crank-shaft; *8* is the fly-wheel; *9* is the eccentric; *10* is the eccentric-strap; *11* is the eccentric rod; *12* is the rocker; *13* is the valve rod or stem, and *V* is the slide-valve. These are all movable parts of the engine, or parts which change their relative positions when the engine is in motion. *W* is the working length of the cylinder. It is slightly less than the distance between the cylinder heads, since a small space must be left between the head and the piston, when the latter is at the end of its stroke. This space, together with the volume of the steam port, which leads to it, is called the **clearance.**

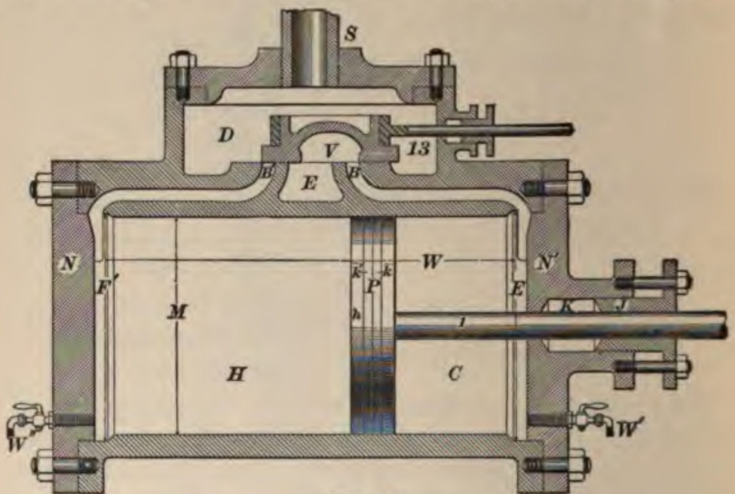


FIG. 243.

1227. The **stroke** of the engine is the travel of the piston *P*; since the piston and cross-head are rigidly fastened to the same rod, the stroke must also be equal to the travel of the cross-head. It was shown in Fig. 241 that the stroke is also equal to the diameter of the circle described by the crank-pin *6*, or, what is the same thing, it is equal to twice the length of the crank *5*, this length being measured from the center of crank-pin *6* to the center of the crank-shaft *7*. M is the diameter or **bore** of the cylinder.

1228. The size of an engine is generally expressed by giving the diameter of the cylinder and the stroke in inches. Thus, an engine having a cylinder diameter of 16 inches, and a stroke of 22 inches, is called a 16″ × 22″ engine.

1229. At the ends F and F' the cylinder is **counter-bored**—that is, for a short distance, the bore is greater than M. The piston projects partly into this counterbore at the end of each stroke. Were it not for the counterbore, the piston would not wear the cylinder walls their entire length, and shoulders would be formed at each end of the cylinder. When the wear of the joints in the connecting-rod is taken up, the length of the connecting-rod is slightly increased, and the piston is shoved back slightly towards the head end of the cylinder. In this case, the shoulder would cause an undesirable pounding of the piston.

Drain-cocks W' and W'' are fitted in each end of the cylinder, through which any condensed steam may be discharged.

1230. The piston fits loosely in the cylinder and has split rings k and k' inserted, which spring out so as to press against the wall of the cylinder, and prevent leakage of steam between the wall of the cylinder and piston. Pistons are usually supplied with a follower-plate h, which is bolted to the head end of the piston P, in order to hold these split rings k and k' in place. The piston rod *1* is a round bar, rigidly connected to both the piston P and the cross-head *2*.

K is a stuffing-box in which packing is placed, and is fitted with a gland J, which, when bolted down, compresses the packing around the piston rod *1*, and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland. The cross-head *2* is given an easy sliding fit between the guide bars G and G', Fig. 242, which are in line with the path of the piston rod, and combine with the cross-head to relieve the piston rod of all bending strains.

The connecting-rod *4* forms the connecting link between the cross-head and crank *5*. The joint between cross-head *2* and connecting-rod *4* is made by the cross-head pin *3*; and the joint between the connecting-rod and crank is made by the crank-pin *6*. Connecting-rods are usually made from 4 to 6 times the length of the crank, or from *4 to 6 cranks* in length, as it is called.

1231. The eccentric *9*, which imparts motion to the slide-valve V, is the exact equivalent of a crank having the same throw. This is clearly shown in Fig. 244, in which *9* is the eccentric, *10* is its strap, *11* is the eccentric rod, and W is a crank having a throw T equal to that of the eccentric. The center of the crank-shaft Q is the center of rotary motion. The dotted circle represents the path of the common center O of the eccentric and crank W. The eccentric revolves freely in the eccentric strap *10*, which is rigidly connected to the eccentric rod *11* (see Figs. 242 and 244). In practice, the diameter of the shaft generally exceeds the throw of the eccentric. In plain slide-valve

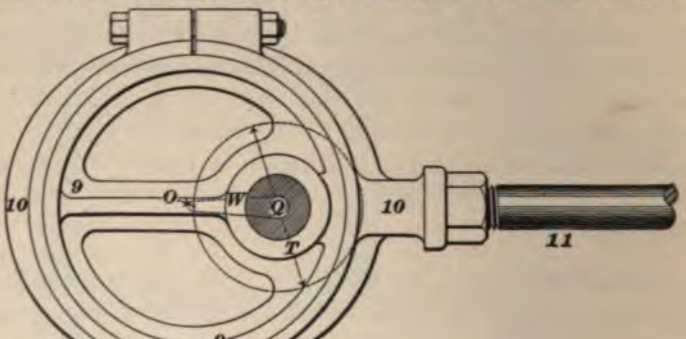


FIG. 244.

engines, the eccentric is usually keyed to the shaft after being properly adjusted. The connection between the eccentric rod *11*, Fig. 242, and the valve stem *13*, is accomplished in a variety of ways. In the figures, a rocker arm *12* is used to support the joint between the eccentric rod *11* and the valve stem *13*. The latter must be supported in some such manner to prevent it from binding in its stuffing-box.

THE D SLIDE-VALVE AND STEAM DISTRIBUTION.

1232. Of the different kinds of valves used to distribute the steam in the engine cylinder, the **D** slide-valve is the most common. A section of such a valve is shown in Fig. 245, in its central position; *p* and *p* are the **steam ports,** *o* and *o* the **bridges,** *E* the **exhaust port,** *S T* the **valve seat.** The flanges of the valve *a b* and *c d* are seen to be wider than the ports which they cover. Of this extra width, the parts *e*, *e* are called the **outside lap,** and the parts *f*, *f* the **inside lap.** The valve is here shown in mid position; i. e., the center line *n* of the valve coincides with the center line *m* of the exhaust port. As the motion of the valve is given by the eccentric, the valve is in mid position when

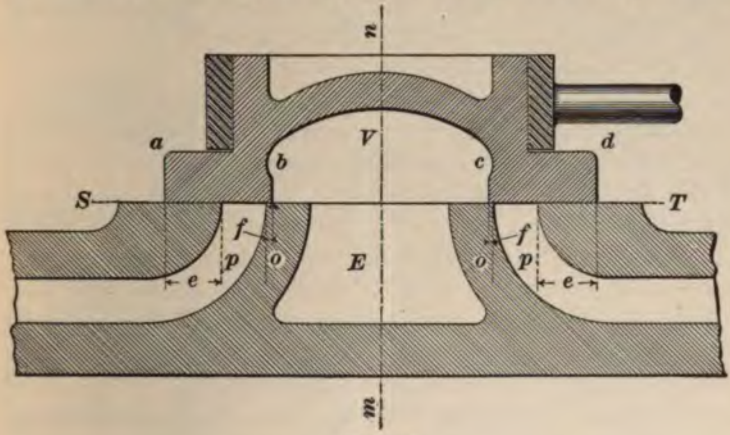


FIG. 245.

the radius of the eccentric (*Q O*, Fig. 244) is in a vertical position. When *Q O* lies horizontally on the right side of *Q*, the valve (see Fig. 242) is in its position nearest the head end of the steam chest, and when *Q O* lies horizontally on the left side of *Q*, the valve is at the end of its stroke in the crank end of the steam chest. In order to show how the steam is distributed in the cylinder by means of the valve and eccentric, a series of skeleton diagrams have been drawn showing the relative positions of the valve and piston for different points of a double stroke.

1233. Fig. 246 shows five diagrams. They represent a **D** slide-valve without lap or lead. *O a* represents the crank; *O b* the eccentric (which was shown to be equivalent to a crank); *a c* the connecting-rod, and *b d* the eccentric rod. It should be remarked that the sizes of the various parts have been greatly exaggerated, particularly the radius of the eccentric circle, and the amount of clearance. Diagram *A*, Fig. 246, represents the piston as just on the point of beginning the forward stroke. The valve is moving in the direction of the arrow, and the outer edge is just about to admit steam to the left-hand port. As will be seen, the valve is in its central position, and, consequently, the line joining the center of the shaft and the center of the eccentric (this line will hereafter be called *eccentric radius*) is vertical. All of the parts are about to move in the direction of the arrows. Diagram *B* shows the position of the parts when the crank has moved through 90° from its position in *A*. The piston is at the middle of its stroke, or very nearly there. It would be exactly at the middle of its stroke, but for the fact that the connecting-rod makes an angle with the horizontal. The angularity of the connecting-rod will be treated of later; for the present, it will be assumed that it has no effect on the position of the piston. The valve has reached the extreme limit of its travel to the right, and the eccentric radius *O b* is horizontal. The left steam port is fully opened for the live steam, and the right steam port is fully opened for exhaust. Another crank movement of 90°

places the different parts as shown in diagram *C*. The piston has reached the end of its forward stroke; the valve is in its central position moving towards the left, and has just

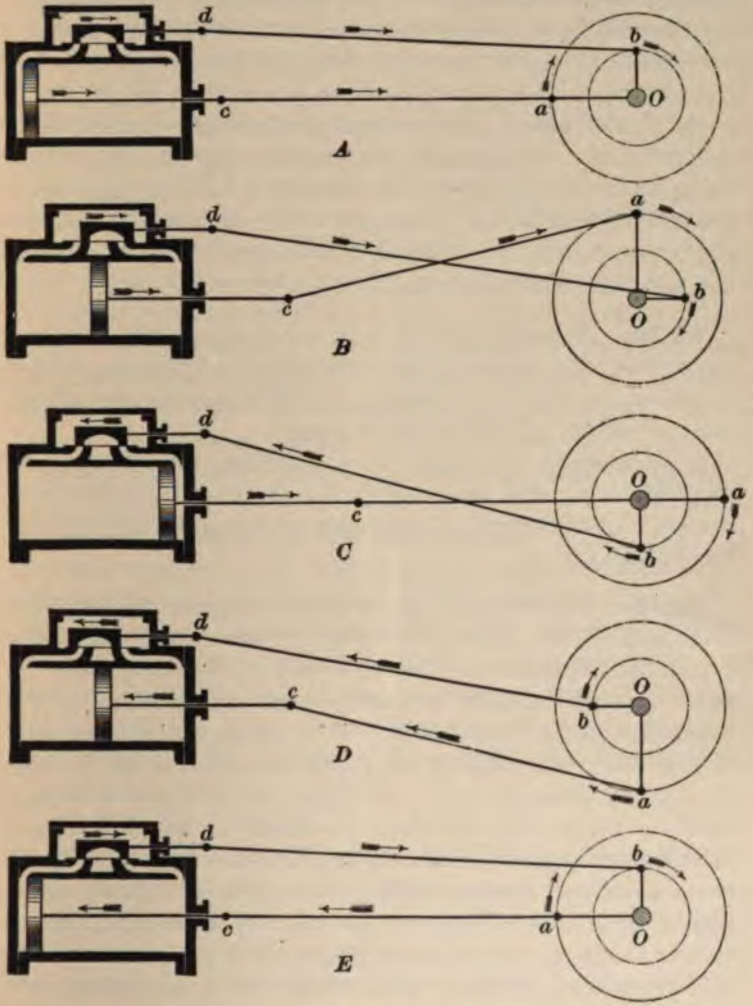


FIG. 246.

closed the left steam port and right exhaust port, and is just about to open the right port for admission of live steam, and the left port for the release of exhaust steam. The piston has now traveled one full stroke. Diagram *D* shows the piston in its central position on the return stroke. The crank is in the position *O a*; the eccentric is horizontal, as represented by *O b*, and the valve is at the farthest point of its travel to the left, the right port being fully open for live steam, and the left port fully open for exhaust. In the diagram *E*, the piston has reached the extreme point of the return stroke, the piston rod, connecting rod, and crank being all in one straight line; this also occurs in diagrams *A* (which is the same as *E*) and *C*. The valve has been moving to the right, and is now in its central position, just on the point of admitting steam to the left port.

1234. These diagrams show conclusively that, with no lap or lead, the steam is admitted to the cylinder for the full stroke of the engine, consequently there can be no cut-off, and, therefore, no expansion of steam.

The following conclusion is now evident: *With an ordinary* D *slide-valve, operated by one eccentric, there can be no cut-off, and, therefore, no expansion of steam, unless the valve has outside lap.*

1235. The effect of lap on the movement of the valve relatively to the piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 247–254. In these figures, the valve has both outside and inside lap, but no lead. These diagrams have been distorted, as was done in Fig. 246, in order that the eccentric radius might be long enough to show up well. In Figs. 247–254 the eccentric radius is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston for the same reason. In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked *M*, and those below, *N*. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side.

Suppose each pipe has a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder, and the upper side to the atmospheric pressure. Suppose, further, that there is a coiled spring on top of the piston; that a piston rod passes through the center of the spring; and that a pencil is attached to the end of the piston rod. If a pressure of 10 pounds is required to compress the spring 1 inch, it is evident that for every 10 pounds pressure in the cylinder, the pencil will move upwards 1 inch, and, if it touched a sheet of paper, would mark a line on that paper. It will now be presumed that an arrangement like that just described is attached to the steam engine piston, and that the pencil touches a sheet of paper, which is held stationary. Then, when the steam piston moves ahead, the pencils will make straight lines at heights corresponding to the steam pressure on the under sides of the little pistons, except when the pressure of the steam in cylinder varies, in which case, the pencil will move up or down, according as the pressure increases or diminishes.

Having made these suppositions clear, let *Q X*, Figs. 247 to 254, represent the line which the pencil would trace if there were a perfect vacuum in the cylinder; i. e., *Q X* is the line of no pressure; also, let *A B* represent the line which the pencil would trace if the pressure in the cylinder was just equal to that of the atmosphere, and *Q Y* the line of no volume. Then the point *Q* represents no volume and no pressure. Finally, let *1 D* represent the volume of the clearance.

1236. Consider Fig. 247. The piston is represented as just beginning the forward stroke, and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at *e*, but on account of the lap, the valve has moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has been moved from *e* to *b*, *O b* being the position of the eccentric radius. The angle *b O e*, which the eccentric radius makes with the position it would be in if there were no lap or lead, is called the **angle of advance.**

Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at full boiler pressure. If the length of the line *A 1* represents the boiler pressure (gauge), the pencil which registers the pressure on the left side of the piston will be at *1*. The steam on the right side of the piston is flowing (*exhausting*) into the atmosphere through the exhaust port as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.** Therefore, in the diagram *N*, let *1* be the position of the second pencil, then, *1 B* is the back pressure.

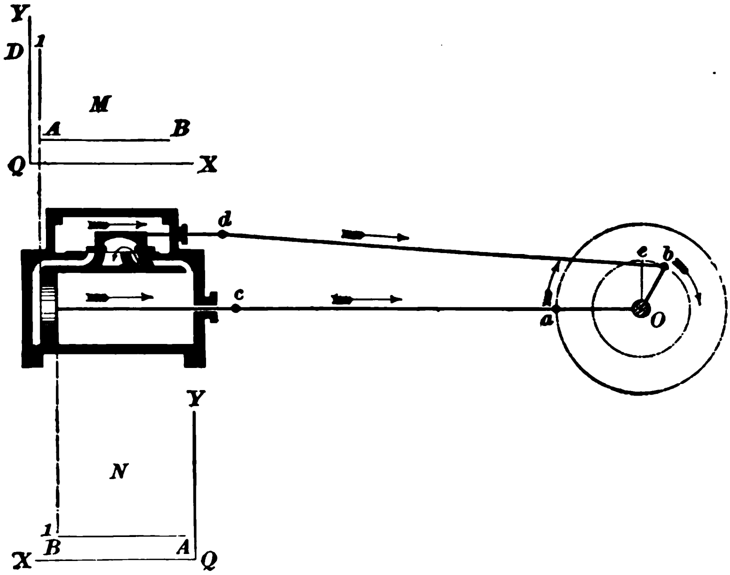


FIG. 247.

1237. Fig. 248 shows the position of the piston and valve when the exhaust port is fully open. The crank has moved from the position $O\ f$ (shown by dotted line) to $O\ a$, and the eccentric center from g to b. Steam is entering the head

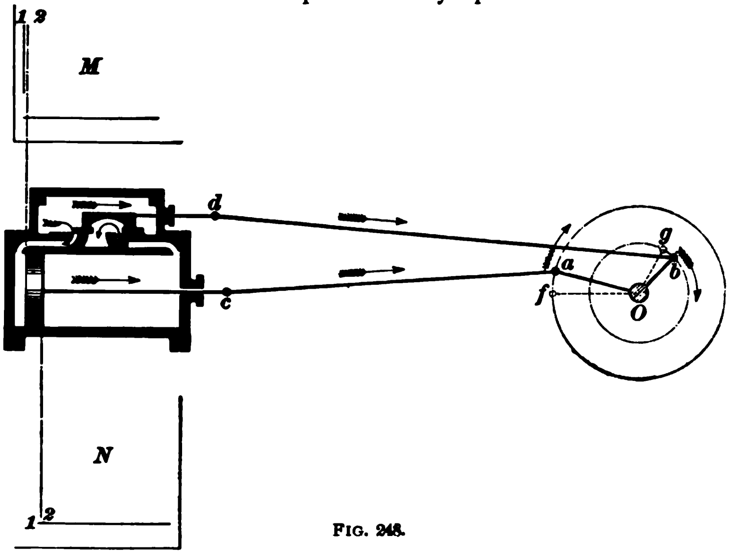


FIG. 248.

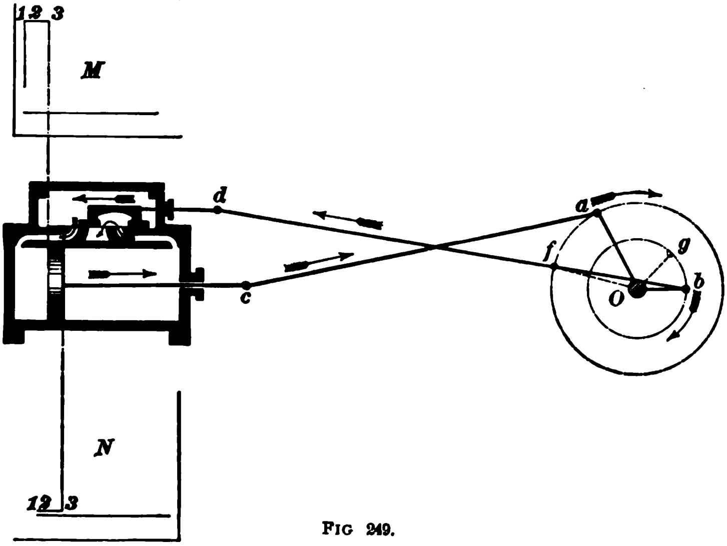


FIG 249.

end of the cylinder and exhausting at the crank end. The pencils have moved from 1 to 2 on both diagrams *M* and *N*.

1238. In Fig. 249 the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*, the dotted lines showing their last position in Fig. 248. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines 2-3.

1239. Fig. 250 shows the piston still further advanced on its stroke, and the valve as having its inside edge in line with the outside edge of the exhaust pipe. The left end of the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines. The pencils have traced the lines *3-4* on the diagrams *M* and *N* during this movement of the piston from the last position.

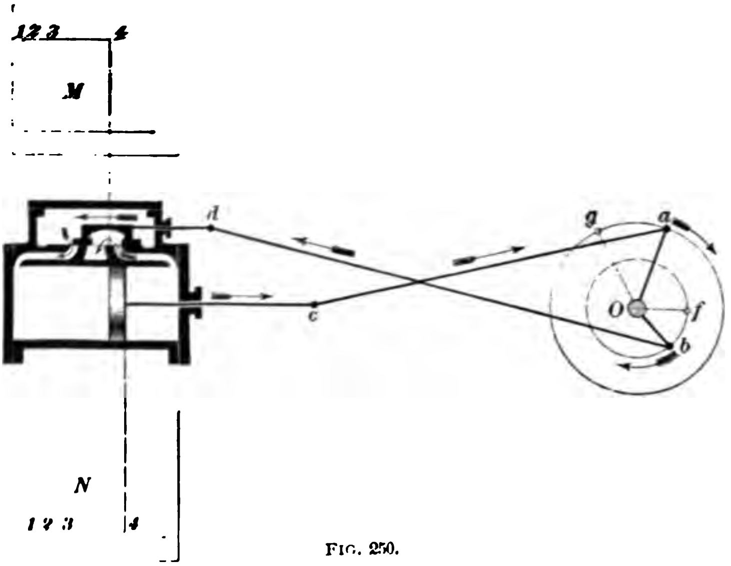


FIG. 250.

Fig. 251 marks one of the most important points of the stroke. Here the valve has closed the steam port; i. e., cut off the steam, and from here to the end of the stroke, the steam in the cylinder expands. The exhaust port is

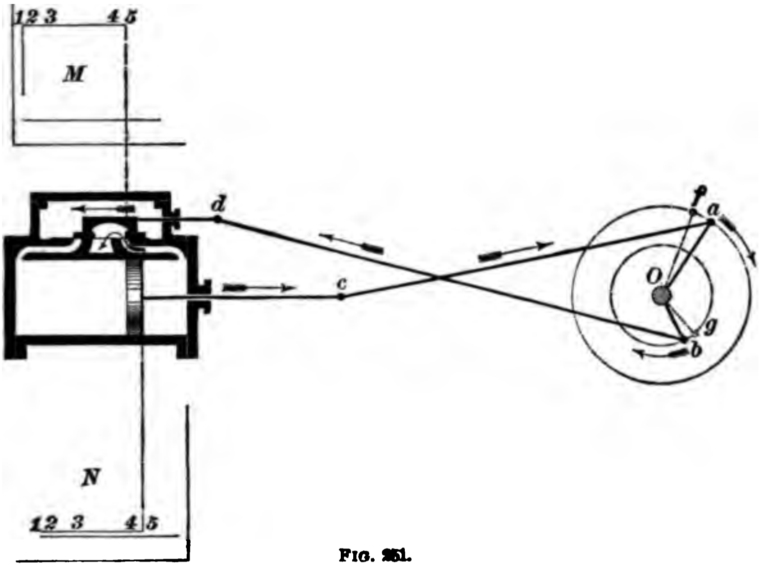


FIG. 251.

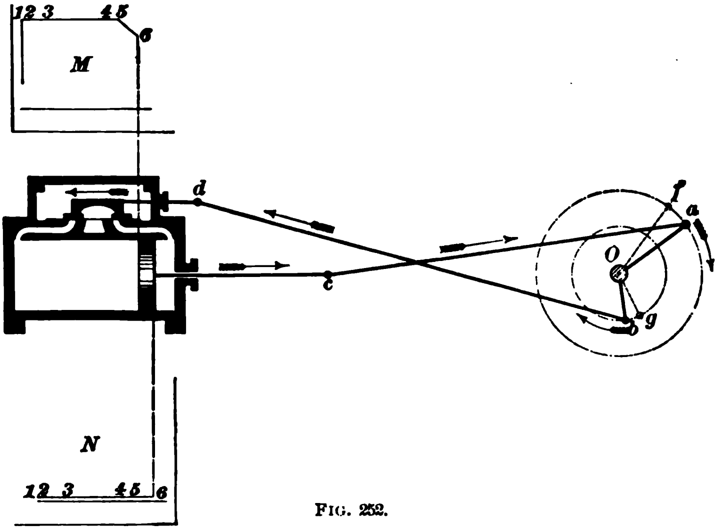


FIG. 252.

now partially closed. The crank and eccentric have moved through the angles indicated. During this movement, the pencils have traced the lines *4-5*.

1240. Fig. 252 shows another very important valve position. Here the inside edge of the valve closes the exhaust port, and, from now on to the end of the stroke, the steam in front of the piston is compressed. In the diagrams *M* and *N*, the lines *5-6* are traced by the pencils. The line *5-6* on the diagram *M* is an expansion line, the pressure falling as the piston moves ahead.

1241. In Fig. 253, the piston has advanced far enough to cause the left inside edge of the valve to be in line with the inside edge of the left port. The slightest movement of the valve to the left will open the left port to exhaust. Expansion really ends here, although, on account of the limitation in the size of the ports, there will still be a slight further expansion owing to the inability of the steam to escape instantly. During this last movement of the

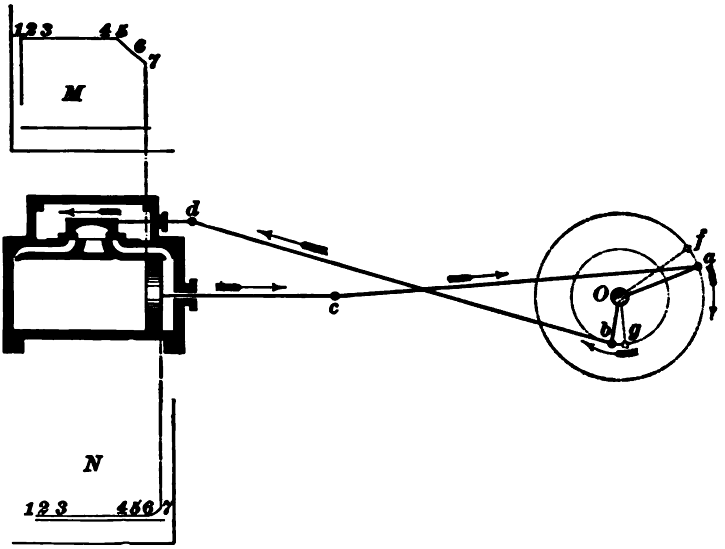


FIG. 253.

piston, the pencils trace the lines *6-7* on the diagrams *M* and *N*. On the diagram *M* the line *6-7* is a continuation of the expansion line *5-6*, while in the diagram *N* it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

1242. In Fig. 254 the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The crank and eccentric are both diametrically opposite their positions in Fig. 247. In the diagrams *M* and *N*, the pencils have traced the lines *7-8*. *M* shows that the pressure has fallen very rapidly from *7* to *8*, while in *N* it has risen from *7* to *8*. The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1'*.

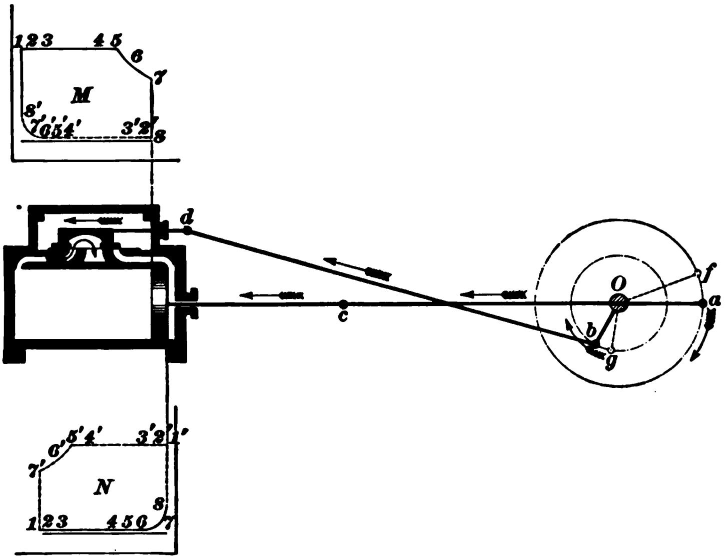


FIG. 254.

1243. During the return stroke the above described actions of the steam will be repeated, the pencils tracing the dotted lines on the diagrams M and N in Fig. 254, the exhaust going through the left port and the steam through the right port. As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions, and note the effects on the diagrams. To assist him in this, the corresponding points have been numbered as in the foregoing figures.

1244. Effects of Lap.—The study of figures 247 to 254 should show the effects caused by varying the lap. Thus, in Fig. 251, it is evident that if the outside lap had been less, the valve would not close the left port when its center was in the position shown; consequently, the piston must move further ahead before the valve can move back far enough to close the port. This, of course, makes the cut-off take place later in the stroke, and shortens the expansion. It is likewise evident that if the valve had more lap, this extra lap would extend beyond the port when the center of the valve was in the position shown. Therefore, the valve would cut off earlier in the stroke and the expansion would be lengthened. Hence, *increasing the outside lap means an earlier cut-off and an increased expansion, while decreasing the outside lap means a later cut-off and a diminished expansion.*

Considering the inside lap, it is evident from Fig. 252 that, if the inside lap had been less, the exhaust port would not have closed so soon, and, consequently, the compression would have begun later; had the inside lap been greater, the compression would have begun earlier. Fig. 253 shows that with a diminished inside lap, the exhaust (usually termed **release**) would begin earlier, while with an increased inside lap, the release would have taken place later in the stroke. Hence, *increasing the inside lap increases compression and delays the release, while diminishing the inside lap decreases compression and hastens release.*

1245. Lead.—A valve is said to have **lead** when it commences to open the steam port just before the piston reaches the end of the stroke. The amount of lead is measured by the distance between the edge of the valve and edge of the port from which the valve is traveling. In Fig. 255, the lead is the distance *a b*. Most engineers give their valves lead in order to have the clearance space filled with steam at boiler pressure when the piston begins its stroke. The effect of lead on the angular advance of the eccentric is evidently the same as an increase of lap; i. e., it increases the angular advance. Its effect on the distribution of steam will be discussed farther on.

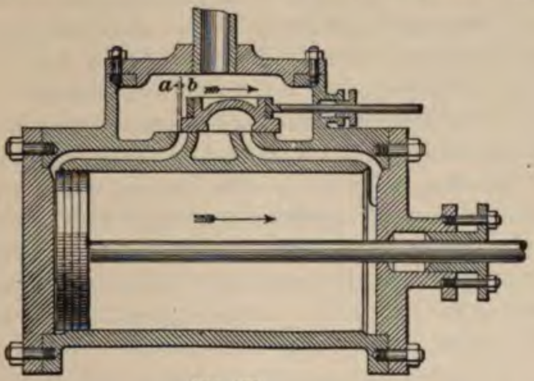


FIG. 255.

1246. In the preceding pages, where the steam distribution has been discussed, it has been assumed that the engine "runs over." When the engine "runs under," the steam distribution and the piston and valve movements will be precisely the same as before, but the position of the center of the eccentric relative to the crank-pin will be changed. To determine the position of the eccentric in this case draw

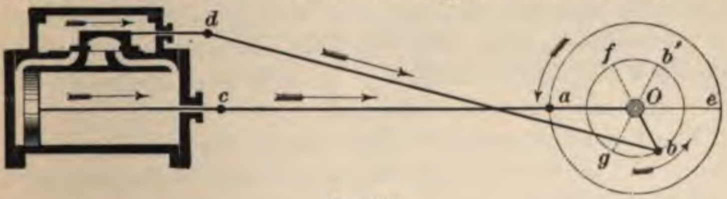


FIG. 256.

the horizontal diameter *a e* of the crank circle, as shown in Fig. 256. *O b'* represents the position of the eccentric radius when the piston is just beginning the forward stroke and the engine runs over. Draw *b' b* perpendicular to *a e* and through the point *b'*; it intersects the valve circle in *b*, and *b* is the position of the center of the eccentric when the engine runs under and is about to begin the forward stroke. It is easy to see that this is so, for the valve and piston must both have a forward movement at this point, whether the engine runs over or under. If the eccentric radius were placed so as to occupy the position *O f*, the forward movement of the piston and downward movement of the crank would cause the valve to move to the left, closing instead of opening the port. It cannot be in the position *O g*, for that would throw the valve too far back. *O b* is the only position in which the eccentric radius can be placed to give the valve the same movement when the engine runs under that it would have if placed in the position *O b'* and the engine ran over. In both cases the valve has the same forward movement, while the center of the eccentric is passing from *b* or *b'* to the horizontal position *O e*.

1247. Rocker Arms.—It frequently happens that the eccentric cannot be so located on the shaft (in the direction of its length) that the eccentric rod and valve stem shall be in the same straight line. It can never be done when the valve is on top of the cylinder without inclining the valve seat, now very seldom done, and, with the valve on the side of the cylinder, other considerations, such as the location of the fly-wheel, may interfere. In such cases as this a lever, or rocker arm, may be employed.

There are four cases which may arise when using a rocker arm: 1. The travel of the valve, the throw of the eccentric, and the direction of motion of the valve and eccentric may be the same as before. 2. The direction of motion of the valve and eccentric may be the same as before, but the travel of the valve may be greater than the throw of the eccentric. 3. The travel of the valve and the throw of the eccentric may be the same as before, but the eccentric may move in the opposite direction. 4. The travel of the valve may be greater than the throw of the eccentric, and the direction of motion may be opposite that in Cases 1 and 2.

The first case is clearly shown in Fig. 242. It is perfectly evident that when the eccentric rod *11* moves to the left that the valve rod *13* will also move to the left, being compelled to do so by reason of the rocker arm *12*. It is also plain that the amount of horizontal movement of the valve rod will also be the same as it would be if the eccentric were attached directly to the valve rod, thus getting rid of the rocker arm. The reason for using the rocker arm in this case is that the eccentric rod axis and valve stem axis are not in the same straight line, the eccentric being then thrown too far to the left by the main bearing R'. The valve seat could, in this case, have been placed further out from the center of the cylinder, so as to bring the axes of the two rods in line. This, however, would have made the steam and exhaust ports that much longer. Since it is considered to be an advantage to have ports as short as possible, a rocker arm was used.

1248. Sometimes the valve travel is such that if the eccentric were made to have the same throw it would be inconveniently large. In such a case the eccentric and its connection to the rocker arm may be left in the same position as shown in Fig. 242, but the valve and its seat may be raised, the valve stem connecting to the rocker arm at a higher point than shown in Fig. 242. The effect of such an arrangement is illustrated in Fig. 257. The direction of motion of valve and eccentric is the same as in Figs. 247 to

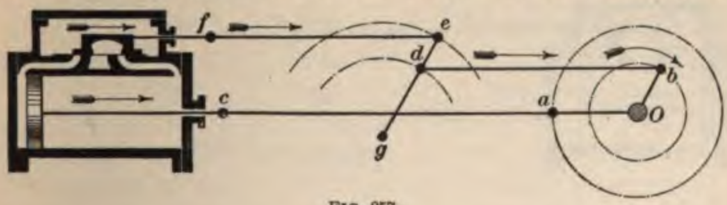


FIG. 257.

254, but the throw of the eccentric is less than the travel of the valve by the ratio $g\,d : g\,e$; i. e., if the valve travel is 4″ and $g\,d = 12''$ and $g\,e = 15''$, the throw of the eccentric will be $4'' \times \frac{12}{15} = 3.2''$. If the engine runs under, the position of the center of the eccentric will be the same as in Fig. 258, and may be found by the same method that was given for finding it in the case shown in Fig. 256.

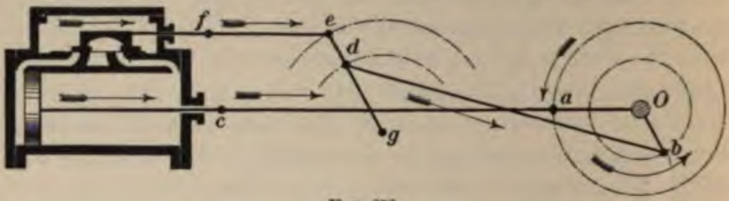


FIG. 258.

1249. In all three of the cases just described the direction of motion of the valve and of the eccentric has remained the same as if there had been no rocker arm, and both points of connection d and e of the valve stem and eccentric rod to the rocker arm were on the *same* side of the pivot g. Suppose that the valve had been placed on the top of the cylinder, and it had been found more convenient to place the pivot of the rocker between the connections of the rocker arm to the valve stem and eccentric rod, as shown in Fig. 259; then, when d moves to the right, along the dotted arc whose center is at g, e moves to the left. Consequently, if the eccentric center was in the position $O\,b'$, and the engine was running in the direction of the arrow, the valve would move backwards instead of ahead. To overcome this difficulty, the eccentric is shifted 180° to the position $O\,b$, then

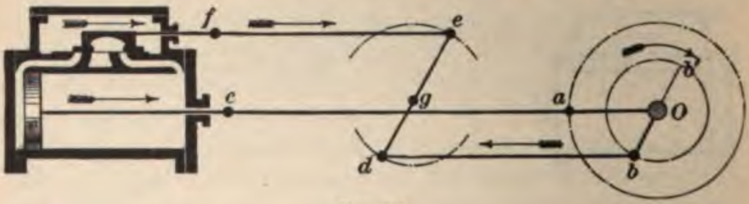


FIG. 259.

a movement of *b* in the direction of the arrow will throw *d* to the left and *e* to the right, as is easily seen.

If *g d* and *g e* are equal, the valve travel and the throw of the eccentric will be equal, fulfilling the conditions for Case 3. If *g d* is less than *g e*, the throw of the eccentric will be less than the valve travel by the ratio *g d* : *g e*. For example, suppose that $g\,e = 20''$ and $g\,d = 15''$ and the valve travel is 5″; then, the throw of the eccentric will be $5'' \times \frac{15}{20} = 3\frac{3}{4}''$.

1250. Fig. 260 shows the position of the eccentric center when the engine runs under, and the rocker arm is of the same design as the one shown in Fig. 259. If there were no rocker arm, the eccentric center would be at *b*′, as in Fig. 256, but since the rocker arm changes the direction of motion in this case, the eccentric is turned around 180°, to a point diametrically opposite.

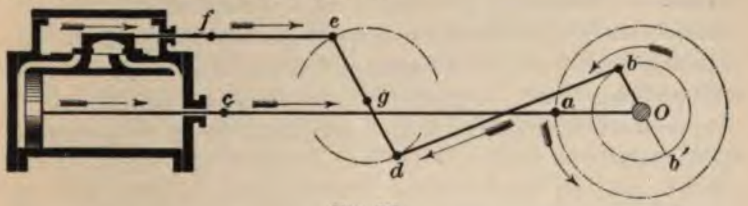


FIG. 260.

The following rule may be applied to any engine whose valves cut off by their outside edges, as has been done in all the previous cases:

Rule.—*Place the crank in the position O a, and the eccentric in the position O b, as shown in Fig. 247, e O b being the angle of advance. If the engine runs over and the rocker arm does not reverse the direction of motion of the eccentric, the eccentric is now correctly set. If the engine runs under, the eccentric should be placed in the position shown in Fig. 256, according to the rule given in connection with that figure. If the engine has a rocker arm whose pivot lies between the point of connection with the valve stem and eccentric rod, and the engine runs over, place the eccentric center diametrically opposite the position shown in Fig. 247. If the engine runs* *under, and the pivot of the rocker arm lies between the two points of connection, place the eccentric center diametrically opposite the position shown in Fig. 256.*

The methods of setting the eccentric, when the valve cuts off by its inside edges, and the determination of the proper amount of lap and lead, will be treated of in Arts. **1621** to **1636.**

1251. Dead Centers.—When the piston has reached the end of either stroke, the piston rod, connecting rod, and crank are all in one straight line, and the entire steam pressure on the piston is transmitted directly to the shaft and bearings, none of it being used to turn the crank. This is clearly shown in Fig. 256. It is evident that there is no turning force on the crank due to the steam pressure when the reciprocating parts are in the position shown. When the crank occupies this position, it is said to be on its **dead center.** There are two dead center positions, $O a$ and $O e$, diametrically opposite each other, corresponding to the two extreme positions of the piston. When the crank occupies the position $O a$ it is said to be on its **interior** dead center, and when it occupies the position $O e$, on its **exterior** dead center.

1252. Clearance.—When the crank is on a dead center, and the piston at the end of its stroke, there is always a space between the piston and the cylinder head. The volume of the space plus the volume of the one *steam* port leading into it is called the **clearance.** Thus, in Fig. 255 the piston is at the end of its return stroke, and the clearance is the volume of the space between the piston and the left cylinder head, plus the volume of the left steam port. In other words, the clearance may be defined as the volume of steam between the valve and the piston, when the latter is at the end of its stroke. The clearance of an engine may be found by putting the engine on a dead center and pouring in water until the space between the piston and the cylinder head, and the steam port leading into it, is filled. The volume of the water poured in is the clearance.

The clearance may be expressed in cubic feet or cubic inches, but it is more convenient to express it as a percentage of the volume swept through by the piston. For example, suppose the clearance volume of a 12″ × 18″ engine is found to be 128 cu. in. The volume swept through by the piston per stroke is $12^2 \times .7854 \times 18 = 2{,}035.8$ cu. in. Then, the clearance is $\frac{128}{2{,}035.8} = .063 = 6.3\%$. The clearance may be as low as ½% in Corliss engines, and as high as 14% in very high speed engines.

1253. Theoretically, there should be no clearance, since the steam which fills the clearance space does no work except during expansion; it is exhausted from the cylinder during the return stroke, and represents so much dead loss. This is remedied, to some extent, by compression. If the compression were carried up to the boiler pressure, there would be very little, if any, loss, since it would then fill the entire clearance space at boiler pressure, and the amount of fresh steam needed would be the volume displaced by the piston up to the point of cut-off, the same as if there were no clearance. It is not practicable to build an engine without any clearance, owing to the formation of water in the cylinder due to the condensation of steam, particularly when starting the engine. As water is practically incompressible, some part of the engine would be broken when the piston reached the end of its stroke, provided there were no clearance space for the water to collect in; usually, the cylinder heads would be blown off. Neither is it practicable to compress to boiler pressure, as a general rule, for that causes too great strains on the engine. Automatic cut-off high-speed engines of the best design, with shaft governors, usually compress to about half the boiler pressure, and have a clearance of from 7% to 14%.

Corliss engines have but very little compression, owing to the construction of the valve; they, likewise, have very little clearance.

1254. Real and Apparent Cut-Off and Ratio of Expansion.—The **apparent cut-off** is the ratio between the portion of the stroke completed by the piston at the point of cut-off, and the total length of stroke. For example, if the length of stroke is 48 inches, and the steam is shut off from the cylinder just as the piston has completed 15 inches of the stroke, the apparent cut-off is $\frac{15}{48} = \frac{5}{16}$.

The **real cut-off** is the ratio between the volume of steam in the cylinder at the point of cut-off, and the volume at the end of the stroke, both volumes including the clearance of the end of the cylinder in question. If the volume of steam in the cylinder (including the clearance) at the point of cut-off is 4 cu. ft., and the volume (including the clearance) at the end of the stroke is 6 cu. ft., the *real* cut-off is $\frac{4}{6} = \frac{2}{3}$.

The **ratio of expansion,** also called the **number of expansions,** is the ratio between the volume of steam (including the steam in the clearance space) at the end of the stroke, and the volume, including the clearance, at the point of cut-off. It is the reciprocal of the real cut-off. For example, if the volume at the end of the stroke is 8 cu. ft., and at cut-off is 5 cu. ft., the ratio of expansion is $\frac{8}{5} = 1.6$; in ordinary language, the steam would be said to have *one and six-tenths expansions.* The corresponding real cut-off would be $\frac{5}{8}$.

Let e = the number of expansions;

i = the clearance, expressed as a per cent. of the stroke;

k = the real cut-off;

k_1 = the apparent cut-off;

r = the apparent number of expansions $= \frac{1}{k_1}$.

$$\text{Then, } e = \frac{1}{k} \text{ and } k = \frac{1}{e}. \qquad \textbf{(96.)}$$

$$k = \frac{k_1 + i}{1 + i}. \qquad \textbf{(97.)}$$

EXAMPLE.—The length of stroke is 36 inches; the steam is cut off when the piston has completed 16 inches of the stroke; the clearance is 4 per cent. Find the apparent cut-off, the real cut-off, and the number of expansions.

SOLUTION.—Apparent cut-off $= \frac{16}{36} = \frac{4}{9} = .444$. Ans.

Real cut-off $= k = \frac{k_1 + i}{1 + i} = \frac{.444 + .04}{1 + .04} = \frac{.484}{1.04} = .465$. Ans.

Number of expansions $= e = \frac{1}{k} = \frac{1}{.465} = 2.15$. Ans.

EXAMPLES FOR PRACTICE.

1. Length of stroke, 18"; apparent cut-off, .4; clearance, 7.5%. Find (*a*) real cut-off, and (*b*) real number of expansions.

Ans. $\begin{cases} (a) & .442. \\ (b) & 2.262. \end{cases}$

2. Length of stroke, 66"; clearance, 4%; steam cuts off at 14¼". Find (*a*) real and (*b*) apparent cut-off in per cent. of stroke, and (*c*) real and (*d*) apparent number of expansions.

Ans. $\begin{cases} (a) & 24.97\%. \\ (b) & 21.97\%. \\ (c) & 4, \text{ nearly}. \\ (d) & 4.552 \text{ nearly}. \end{cases}$

INDICATORS AND INDICATOR CARDS.

1255. The arrangement described in connection with Figs. 247 to 254, for recording the steam pressure at all points of the stroke of the piston, would be impossible to put into actual operation. Again, the diagram traced by the pencil would be altogether too large to be handled conveniently. To overcome these objections, an instrument called the **indicator** is used. The principal reason for obtaining a diagram of this kind is that it affords a ready means of computing the mean pressure of the steam upon the piston during one stroke. If the mean pressure on both sides of the piston, the length of the stroke, and the number of strokes per minute are known, the horsepower of the engine can be easily found.

1256. Fig. 261 shows the general appearance of an indicator, and Fig. 262 shows one in section. The instrument consists essentially of a cylinder *a* containing the piston *g* and spiral spring *d*. By turning a cock connected to the small pipe to which the indicator is attached, steam may be admitted to, or shut off from, the cylinder of the indicator at pleasure. When steam is admitted through the channel *s*, Fig. 262, its pressure causes the piston *g* to rise. The spiral spring *d* is compressed, and resists the upward movement of the piston. The height to which the piston rises should then be in exact proportion to the pressure of the steam, and, as the steam pressure rises and falls, the piston must rise and fall accordingly.

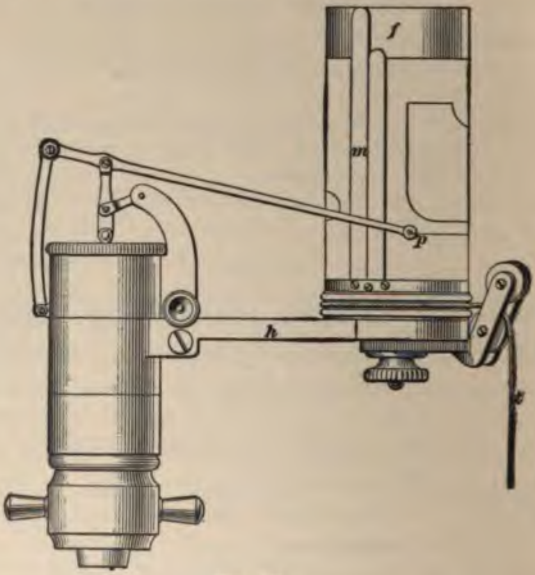


FIG. 261.

To register this pressure, a pencil might simply be attached to the end of the piston rod *e*, the point of the pencil being made to press against a piece of paper. It is desirable, however, to restrict the maximum travel of the piston to about half an inch; while the height of the card may advantageously be two inches or more. To obtain a long range of the pencil, combined with a short travel of the piston, the pencil is attached at *p* to the long end of the lever *n k p*. The fulcrum of the lever is at *n*. The piston rod is connected to it at *k* through the link *i*. The pencil motion is thus $\frac{pn}{nk}$ times the piston travel. This ratio $\frac{pn}{nk}$ is, for most indicators, either 4, 5, or 6. The point *p* is forced to move in a vertical straight line by the arrangement of the links and joints *i*, *c*, *n*, *k*.

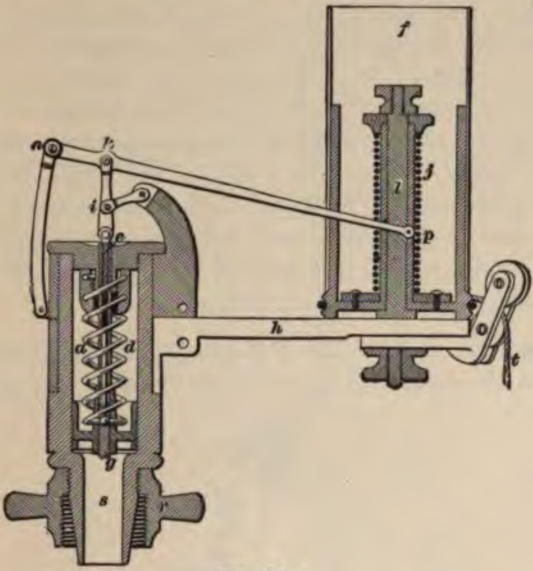


FIG. 262.

1257. The height to which the piston will rise under a given steam pressure depends upon the stiffness of the spring. Indicators are usually furnished with a number of springs of varying degrees of stiffness, which are distinguished by the numbers 20, 30, 40, etc.

These numbers indicate the pressure per square inch required to raise the pencil one inch. Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil one inch, and, therefore, the vertical scale of the diagram is 40 pounds per inch. That is, the vertical distance in inches of any point on the diagram from the atmospheric line, multiplied by 40, gives the gauge pressure per square inch at that point. The scale of the spring chosen should not be less than half the boiler pressure. For example, we would choose a 40 spring for a steam pressure of 75 pounds per square inch.

1258. The indicator, however, must not only register pressures, but it must register them in relation to the position of the piston. To accomplish this object, a cylindrical drum *f* is provided. This drum can be revolved on its axis *l*, by pulling the cord *t*, which is coiled around it. When the pull is released, the spring *j* rotates the drum back to its original position. If now the cord *t* be attached to some part of the engine which has a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

1259. To attach the indicator to the engine, a hole is drilled in the clearance space of the cylinder and tapped for a ½″ nipple. The nipple has an elbow, into which is screwed the indicator cock. The indicator is then attached directly to the cock by the nut *r*, the conical projection *s* of the indicator wedging tightly into the cock to prevent the

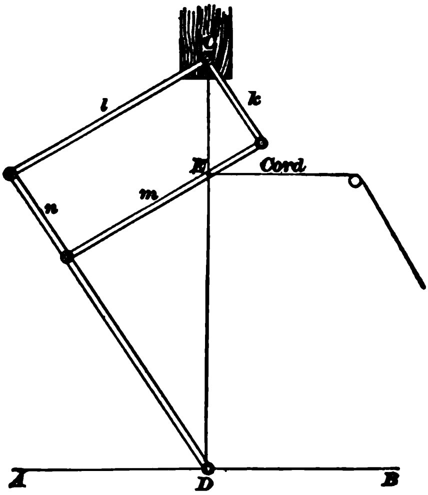


FIG. 263.

leakage of steam. It is preferable to have an indicator at each end of the cylinder, but if that is not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock, as shown in Fig. 264.

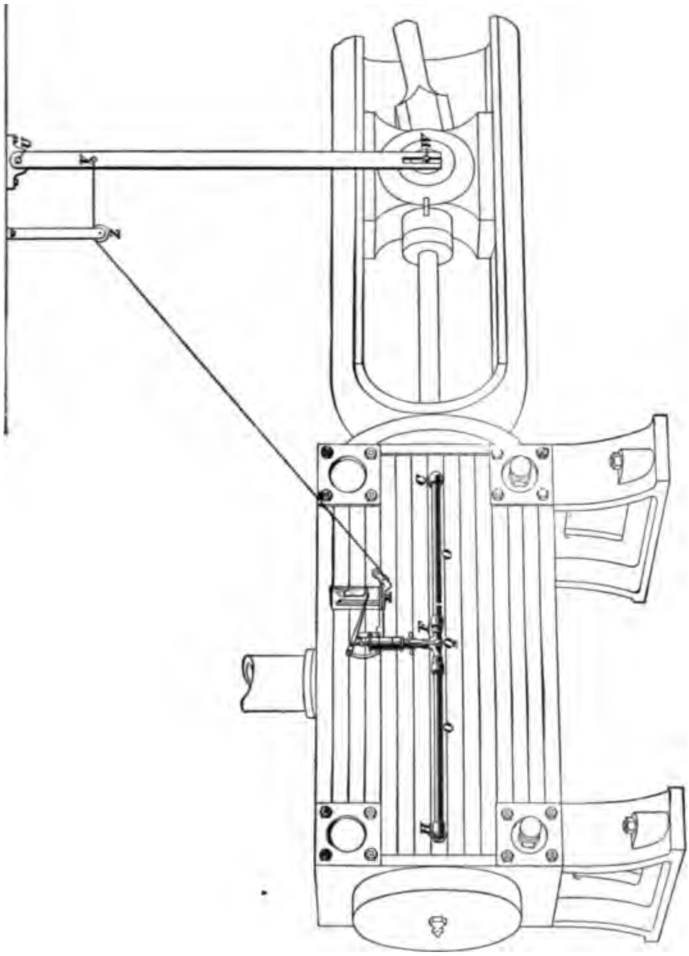


FIG. 264.

The motion of the drum cord is usually obtained from the cross-head. Since the stroke of the engine is nearly always greater than the circumference of the drum, the cord cannot be attached directly to the cross-head, and an arrangement called a **reducing motion** is resorted to. A common form of reducing motion is shown in Fig. 263. It consists of four links joined together in the form of a parallelogram. One of the links n is prolonged, and is pivoted at the end to the cross-head D. The opposite corner of the parallelogram is pivoted to the fixed point C. The cord is attached to the point E on the link m, which point must be on the straight line connecting C and D. $A\,B$ represents the length of the stroke. Letting L represent the length of the indicator diagram, we have the following proportions: $A\,B : L = C\,D : C\,E$, or $\frac{A\,B}{L} = \frac{C\,D}{C\,E}$. This reducing motion is called the pantograph.

1260. Another reducing motion is shown in the "slotted swinging lever," shown in Fig. 264. A pin in the cross-head moves in a slot in one end of the lever, the other end of which is pivoted, at U, to some stationary body. The cord is attached at V. As in the former arrangement,

$$\frac{\text{length of diagram}}{\text{length of stroke}} = \frac{U\,V}{U\,W}, \text{ or}$$

$$U\,V = \frac{\text{length of diagram} \times U\,W}{\text{length of stroke}}.$$

Therefore, to find the distance $U\,V$ between the pivot U and the point V where the cord is attached, multiply the length of the lever by the desired length of diagram, and divide the product by the length of stroke, all the dimensions being taken in inches.

EXAMPLE.—The stroke of an engine is 28 inches; the length of the slotted lever is 6 feet. How far from the pivot must the string be attached to give a diagram 3½ inches long?

SOLUTION.— $\frac{72 \times 3\frac{1}{2}}{28} = 9$ inches.

TO TAKE THE CARD.

1261. The instrument being attached to the engine, as explained above, slip a blank card over the drum, as shown in Fig. 261. Fasten the cord to the reducing gear, taking care that it is not too short, and that the drum does not "strike" at either end of the stroke. (For convenience, the cord should be divided, one part being attached to the reducing gear, and the other to the drum. One end is provided with a loop and the other with a hook, so that they may be connected and disconnected easily.) Open the cock and let the engine make a few strokes to warm up the indicator; then press the pencil gently against the rotating drum during one revolution. Shut the cock and again press the pencil against the drum, to obtain the atmospheric line. Now, disconnect the drum cord and take off the card.

If but one indicator and a three-way cock are used, as shown in Fig. 264, open the cock to admit steam from one end of the cylinder, and take the card from that end; then turn the cock to admit steam from the other end, and take that card; then shut off the steam entirely, and take the atmospheric line.

1262. Figs. 265 and 266 are cards taken from the head and crank ends of the cylinder, respectively. The different phases during the stroke are very clearly shown.

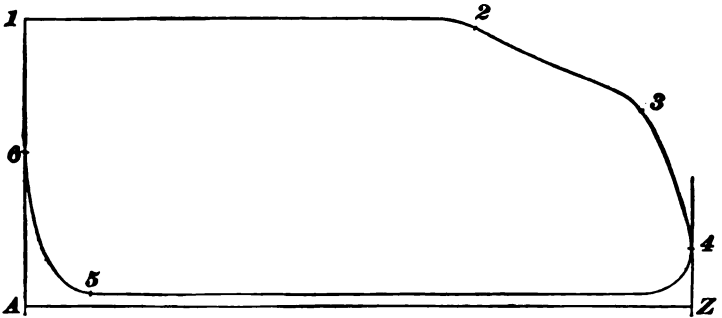


FIG. 265.

Thus: *1* is the beginning of the stroke;
2 is the point of cut-off;
3 is the point of release;
4 is the end of the stroke;
5 is the point of compression;
6 is the point of admission.

The lines included between any two of these points have received special names, which are as follows:

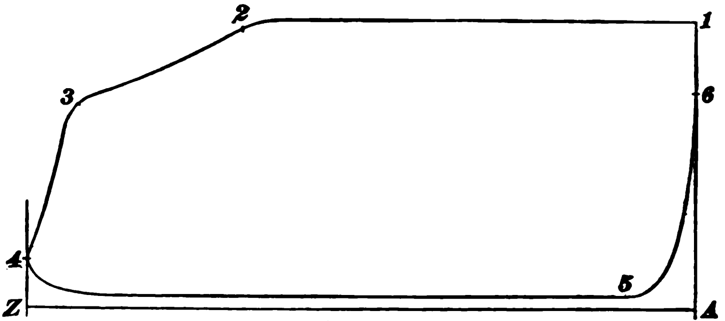


FIG. 266.

6-1 is the admission line;
1-2 is the steam line;
2-3 is the expansion curve;
3-4-5 is the period of release;
4-5 is the back pressure line;
5-6 is the compression curve;
A Z is the atmospheric line.

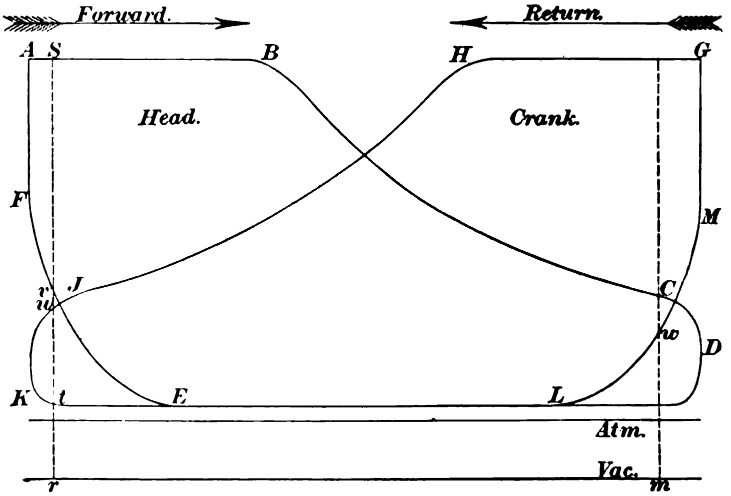


FIG. 267.

1263. If but one indicator is used, the two diagrams may be taken on the same blank as shown in Fig 267.

With the diagrams placed one over the other, as shown, it is very easy to tell exactly what is taking place in the cylinder at any point of the stroke. On the forward stroke the pencil of the indicator describes the line *A B C D* of the head diagram, if the cock is opened to the head end, or it describes the line *K L M*, if the cock is opened to the crank end. Likewise, the lines *G H J K* and *D E F* are described during the return stroke.

Suppose the piston is at a position corresponding to *r* on the forward stroke; the pressure (absolute) urging the piston forward is *r S*, while the pressure resisting is *r t*. Hence, the net pressure on the piston is *S t*. Suppose, now, that the piston is at *r* on the return stroke; the pressure at the right urging the piston on is *r u*, while the pressure on the left is *r v*. The net pressure is, therefore, *u v*, and is negative; or, in other words, the resistance is greater than the effort.

1264. A double diagram of this character tells at a glance what is taking place at either end of the cylinder at any point of the stroke. Thus, when the piston is on the forward stroke, in the position corresponding to *m*, the steam in the head end is at the point of release, as shown at *C*. Draw a line through *m* perpendicular to the vacuum line. *C* lies on *A B C*, and since *K L M* is described at the same time as *A B C*, the intersection of the line through *C* with the line *K L M* is the point corresponding to *C*. Since *w* is on the compression line, compression is taking place in the crank end when release occurs in the head end.

HORSEPOWER.

1265. In order to find from the diagram the horsepower exerted by the engine, we must first find the *mean effective pressure*.

The **mean effective pressure,** or M. E. P., is defined as the average pressure urging the piston forward during its entire stroke in one direction, less the pressure that resists its progress.

The mean effective pressure may be found in two ways: 1. The area of the diagram in square inches may be found by an instrument called the planimeter; the M. E. P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches, and multiplying by the scale of the spring.

Example.—The area of the diagram is 4.2 sq. in., and the length is 3.5 in.; a 40 spring being used, find the M. E. P.

Solution.—$\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in. M. E. P. Ans.

2. Where a planimeter is not available, the following method of finding the M. E. P. is fairly rapid and accurate: Draw tangents to each end of the diagram perpendicular to the atmospheric line. Divide the horizontal distance between the tangents into 10 or more equal parts (10 or 20 parts are the most convenient, but any other number may be used). Indicate by a dot on the card the center of each division, and draw lines through these dots parallel to the tangents from the upper line to the lower line of the card. On a strip of paper mark off successively the length of these lines, the total length thus representing the sum of all the lines. Divide this total length by the number of the lines used, and multiply the quotient by the scale of the spring. The result will be the M. E. P. This method is the same as that given in Art. **1159.**

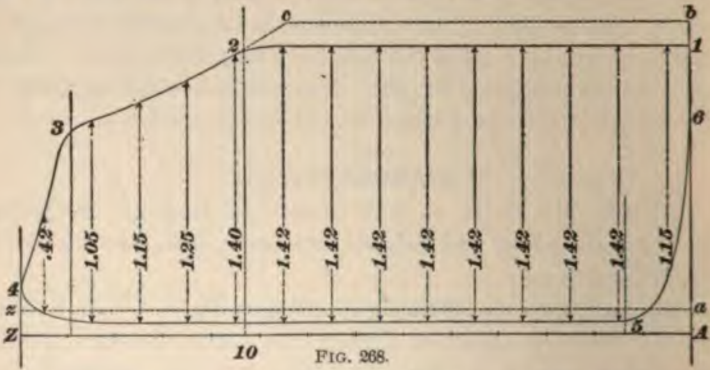


Fig. 268.

Example.—The projection of the head end card of Fig. 265 (see Fig. 268) upon the atmospheric line is the distance *A Z*, and it is divided, in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the card opposite the centers of these spaces, is marked on the lines themselves, and the sum of these lengths is 18.11 inches. The scale of the spring used in obtaining the card was 40 pounds; therefore,

$$\frac{18.11}{14} \times 40 = 51.74 \text{ pounds per square inch} =$$

the M. E. P. of the "head" end card.

Example.—The projection of the crank card of Fig. 266 (see Fig. 269), upon the atmospheric line, is the distance *A Z*, and it is divided in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the card opposite the centers of these spaces, is marked on the lines themselves, and the sum of these lengths is 17.78 inches. The scale of the spring is 40 pounds; therefore,

$$\frac{17.78}{14} \times 40 = 50.8 \text{ pounds per square inch} =$$

the M. E. P. of the "crank" end card.

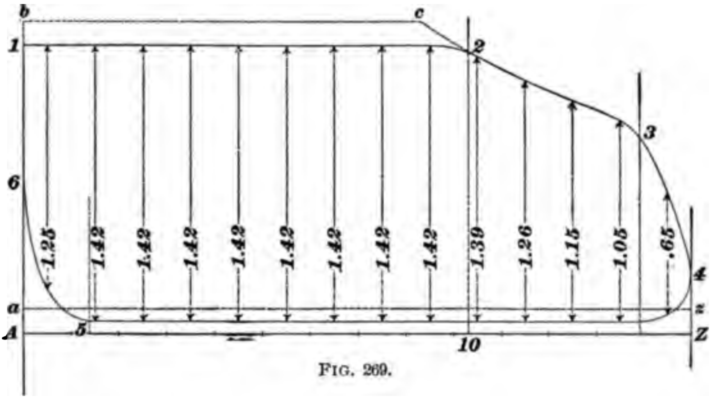


FIG. 269.

Therefore, the M. E. P. in the cylinder during a complete revolution of the crank is

$$\frac{51.74 + 50.8}{2} = 51.27 \text{ pounds per square inch.}$$

1266. The reason for dividing the diagram into 10 parts instead of some other number, is that it shortens the work of calculation. Thus, in the two examples just given, if the number of divisors had been 10 instead of 14, and the sum of the ordinates had been 12.94″, the mean ordinate would have been $\frac{12.94}{10} = 1.294$, and the M. E P., $1.294 \times 40 = 51.76$ lb. per sq. in. All that is necessary is to add the ordinates and shift the decimal point one place to the left to obtain the mean ordinate when the diagram is divided into 10 equal parts. This method saves the time required to divide by some inconvenient number, as 14.

1267. Sometimes the expansion line of the card will fall below the back pressure line, as shown in Fig. 270. In such a case the area of the loop *C D* must be subtracted from the remainder of the card *A B C*. When the planimeter is used, the subtraction is made automatically by the instrument; but when the card is divided into parts by the method of ordinates the sum of the ordinates of *C D* must be subtracted from the sum of those of *A B C A*. The result divided by the number of spaces will give the mean ordinate; multiplying this by the scale of the spring will give the M. E. P., as before.

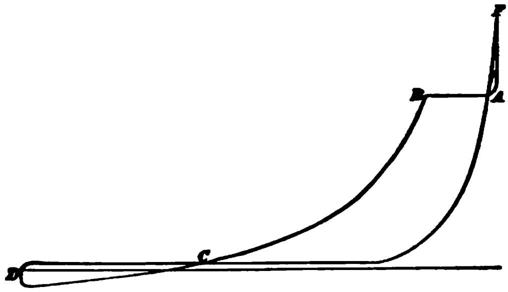


FIG. 270.

1268. We have now all the materials for finding the work done in the engine cylinder, expressed in horsepowers.

Work is the product of force into the distance through which the force moves. In the case of the engine cylinder the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance moved through in a minute is the number of strokes in a given time multiplied by the length of stroke.

Let P represent the M. E. P. in lb. per sq. in.;
A represent the area of piston in sq. in.;
L represent the length of stroke in ft.;
N represent the number of strokes per min.

Then, the work done per minute is $P\,L\,A\,N$ ft.-lb.

One horsepower = 33,000 ft.-lb. per min.

Therefore, the indicated horsepower of the engine is found from the formula,

$$\text{I. H. P.} = \frac{P L A N}{33{,}000}. \qquad \textbf{(98.)}$$

1269. When the point of real cut-off and the steam pressure at the beginning of the stroke are known, the M. E. P. may be found approximately by the following formula:

$$\text{M. E. P.} = \frac{.9\,P(1 + 2.3 \log e)}{e} - .9\,p, \qquad \textbf{(99.)}$$

in which P = absolute steam pressure = gauge pressure + 14.7 pounds;

e = ratio of expansion;

p = absolute back pressure.

p is usually taken as about 3 pounds for condensing engines, and 17 pounds for non-condensing engines.

Example.—A non-condensing engine cuts off at $\frac{2}{3}$ stroke. The clearance is 5%; the gauge pressure is 59.3 pounds. What is the approximate M. E. P.?

Solution.—$k_1 = \frac{2}{3}$. $k = \frac{k_1 + i}{1 + i} = \frac{.66\frac{2}{3} + .05}{1.00 + .05} = \frac{.71\frac{2}{3}}{1.05} = \frac{215}{315} = \frac{43}{63}$. $e = \frac{1}{k} = \frac{63}{43}$, $\log \frac{63}{43} = .16587$. $P = 59.3 + 14.7 = 74$ lb. M. E. P. $= \frac{.9\,P(1 + 2.3 \log e)}{e} - .9\,p = \frac{.9 \times 74 \times (1 + 2.3 \times .16587)}{\frac{63}{43}} - .9 \times 17 =$ 47.5 lb.

Example.—The diameter of the piston of an engine is 10 inches, and the length of stroke 15 inches; it makes 250 revolutions per minute, with a M. E. P. of 40 pounds per square inch; what is the horsepower?

Solution.—As it is not stated whether the engine is single or double acting, assume that it is double-acting. Then, the number of strokes is $250 \times 2 = 500$ per minute. Substituting in formula **98**,

$$\text{H. P.} = \frac{P L A N}{33{,}000} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P.}$$

1270. The product $L\,N$ of the above formula gives the total distance in feet traveled by the piston per minute. It is called the **piston speed.** If the length of stroke L be taken in inches, the piston speed becomes $\frac{LN}{12}$. If R = number of revolutions per minute, $\frac{LN}{12} = \frac{L \times 2R}{12} = \frac{LR}{6}$.

Letting S represent the piston speed in feet per minute,

$$\left.\begin{aligned} S &= \frac{LR}{6} \\ L &= \frac{6S}{R} \\ R &= \frac{6S}{L} \end{aligned}\right\} \quad \textbf{(100.)}$$

The piston speeds used in modern practice are about as follows:

	ft. per min.
Small stationary engines, -	300 to 600.
Large stationary engines,	600 to 1,000.
Corliss engines, - - - -	400 to 750.
Locomotive engines, - -	600 to 1,200.

1271. Having given the I. H. P. of the engine and knowing the available M. E. P., there are two methods of calculating the length of stroke and diameter of piston:

1. We may assume the number of revolutions, and the ratio of the length of stroke to the diameter of cylinder.

2. We may assume a suitable piston speed, and choose the number of revolutions and length of stroke to correspond.

An example will serve to illustrate the above methods.

Given an engine which is to develop 250 I. H. P. with a M. E. P. of 50 pounds per square inch. Find the diameter of piston and length of stroke.

First, let us assume that the engine makes a certain number of revolutions per minute, as 75, and that the length of stroke in *inches* is say twice the diameter of the piston. Substituting in formula **98**,

$$250 = \frac{50 \times L \times A \times (75 \times 2)}{33{,}000},$$

$$\text{or } LA = \frac{33{,}000 \times 250}{50 \times 75 \times 2} = 1{,}100.$$

In the above expression, L is taken in *feet*. It is more convenient to use inches, so we will multiply both sides of the equation by 12.

$$12\,L_{ft.} \times A = L_{in.} \times A = 13{,}200.$$

But $A = .7854\,D^2$ and $L = 2\,D$, according to the above assumption.

Substituting, $L\,A = 2\,D \times .7854\,D^2 = 1.5708\,D^3 = 13{,}200.$

$$D^3 = 8{,}408.$$

$$D = \sqrt[3]{8{,}408} = 20.33 \text{ in.}$$

$$L = 20.33 \times 2 = 40.66 \text{ in.}$$

Second method. We may assume a certain piston speed, say 500 feet per minute; then, as above,

$$250 = \frac{50 \times L\,A\,N}{33{,}000}.$$

But the piston speed $= 500$ feet $= L\,N$.

$$\text{Therefore, } 250 = \frac{50 \times A \times 500}{33{,}000},$$

$$\text{or } A = \frac{250 \times 33{,}000}{50 \times 500} = 330 \text{ sq. in.}$$

$$.7854\,D^2 = 330$$

$$D^2 = 420$$

$$D = 20\tfrac{1}{2} \text{ inches.}$$

If we assume, as before, 75 revolutions per minute, the length of the stroke, from formula **100**, is

$$\frac{500 \times 6}{75} = 40 \text{ inches.}$$

EXAMPLES FOR PRACTICE.

1. The mean ordinates of two diagrams taken from each end of the cylinder of an 18″ × 20″ non-condensing engine running at 200 R. P. M. (revolutions per minute) are respectively .72″ and .76″ long. The scale of spring being 80, what is the horsepower of the engine?

Ans. 304.885 H. P.

2. In the above engine, assume the initial pressure to be 118 lb. per sq. in., gauge; the apparent cut-off as ¼, and the clearance as 8%. (*a*) Find the theoretical M. E. P., and (*b*) the horsepower.

Ans. { (*a*) 64.41 lb. per sq. in.
(*b*) 331.12 H. P.

3. An engine running at 165 R. P. M. has a stroke of 28"; what is the piston speed? Ans. 770 ft. per min.

4. If an engine has a piston speed of 960 ft. per min., and runs at 72 R. P. M., what is the length of the stroke? Ans. 80 in.

5. Initial pressure, 82 lb., gauge; number of expansions, 1.83; back pressure, 4.2 lb., absolute; what is the theoretical M. E. P.? Ans. 72.484 lb. per sq. in.

6. I. H. P., 536.42; piston speed, 480 ft. per min.; M. E. P., 61.15 lb. per sq. in. Find diameter of cylinder to nearest ¼". Ans. 27¾ in.

7. A 16" × 20" engine develops 138 I. H. P. with 35 lb. M. E. P.; how many R. P. M. does it make? Ans. 194.2 R. P. M.

8. A 54" × 66" Porter-Allen non-condensing engine develops 1,382.4 I. H. P., with an initial pressure of 63 lb., gauge, when cutting off at ¼ stroke and running at 82 R. P. M. (*a*) What is the M. E. P.? (*b*) With a back pressure of 1 lb. above the atmosphere and a clearance of 3%, what would be the theoretical I. H. P. calculated by formulas 98 and 99?

Ans. { (*a*) 22.068 lb. per sq. in.
{ (*b*) 1,558.1 I. H. P.

9. A 16" × 14" engine runs at 240 R. P. M.; what is the piston speed? Ans. 560 ft. per min.

10. If the average M. E. P. of the engine in the last example is 41.73 lb. per sq. in. and the diameter of the piston rod is 4", what is the I. H. P., taking the piston rod into consideration? Ans. 187.932 H. P.

1272. From the measurement of the indicator diagrams has been obtained what we have termed the indicated horsepower, or I. H. P.—that is, the total horsepower developed in the engine cylinder. One portion of the I. H. P. is absorbed in overcoming the friction of the engine itself. The remainder is available for doing the required work.

The power absorbed by the engine itself is termed **friction horsepower.**

The power available for doing useful work is termed the **net** or **actual horsepower.**

1273. The actual horsepower of any engine is found by first computing its I. H. P. from a set of indicator cards taken when the engine is running under full load, and then subtracting from this the I. H. P. computed from a set of indicator cards taken when the engine is running under no load, but making the same number of revolutions per minute as above. The horsepower developed by the engine in this last case will only be sufficient to keep the working parts of the engine in motion at the same speed. To produce this result, some means will have to be resorted to of checking the steam supply. These will be discussed later.

EXAMPLE.—Indicator cards taken from an engine when running under full load, and having a piston speed of 498 feet per minute, showed an *indicated horsepower* of 242.7. With the same piston speed, and running under no load, the indicator cards showed an *indicated horsepower* of 75.2. Therefore, $242.7 - 75.2 = 167.5$, which is the *actual horsepower* of the engine.

1274. The **mechanical efficiency** of an engine is the ratio of the *actual horsepower* to the *indicated horsepower;* or it is the per cent. of the mechanical energy developed in the cylinder which is utilized in the doing of useful work.

To find the efficiency of an engine, when the *indicated* and *actual horsepowers* are known :

Rule.—Divide the *actual horsepower* by the *indicated horsepower.*

EXAMPLE.—The indicated horsepower of an engine is 242.7, and the actual horsepower is 197.5. Therefore, $\frac{197.5}{242.7} = 81.38$ per cent. efficiency.

The mechanical efficiency of engines in good order varies from 75 to 90 per cent.

1275. The **thermal efficiency** of the engine is the same as that of any other heat engine. This was shown in Art. **1182** to be $\frac{T_1 - T_2}{T_1}$, where T_1 is the absolute temperature of the entering steam, and T_2 the absolute temperature of the exhaust steam.

EXAMPLE.—The pressure of the entering steam is 100 pounds above vacuum, and at exhaust it is 16 pounds above vacuum; what is the thermal efficiency ?

SOLUTION.—

Temperature of incoming steam, from table, 327.625°.
" " exhaust " " " 216.347°

Absolute $T_1 = 327.625 + 460 = 787.625$.
" $T_2 = 216.347 + 460 = 676.347$.

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1} = \frac{787.625 - 676.347}{787.625} = 14.13 \text{ per cent.}$$

READING INDICATOR DIAGRAMS.

1276. The determination of the I. H. P. is not the only or the most important function of the indicator. By its use, defects in steam distribution may be detected, the correction of which may result in largely increased economy in the working of the engine.

The form of a good diagram depends largely upon the type of engine, style of valve, and speed. The same style of diagram is not possible or desirable from all engines.

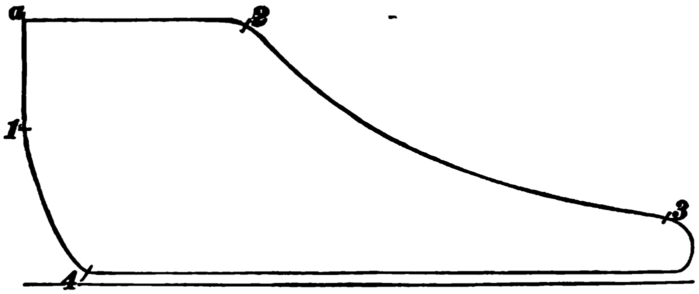


Fig. 271.

Some of the most common faults in steam distribution are given below :

In Fig. 271, *1* is the admission.
2 " " cut-off.
3 " " release.
4 " " compression.

I. Admission may be too early.
II. " " " " late.
III. Cut-off " " " early.
IV. " " " " " late.
V. Release " " " early.
VI. " " " " late.
VII. Compression " " " early.
VIII. " " " " late.

1277. Case I.—The effect upon the diagram of a too early admission is shown in Fig. 272. It is seen that the admission line *1a* instead of being straight and perpendicular to the atmospheric line, as in Fig. 271, curves backwards. With a single valve, like the one previously described, all of the other events, cut-off, release and compression, are also too early. The remedy is to decrease the angular advance of the eccentric.

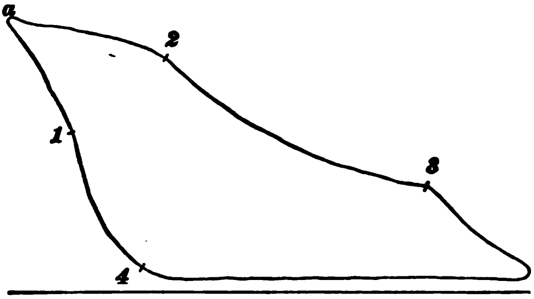


FIG. 272.

1278. Case II.—In this case the admission is too late, and the admission line *1 a*, on a diagram, will curve forwards, as shown in Fig. 273. The remedy is to increase the angular advance until the admission line *1 a* becomes perpendicular to the atmospheric line. It will be noticed that, in the case of a too late admission, the other events at *2*, and particularly *3* and *4*, are also too late.

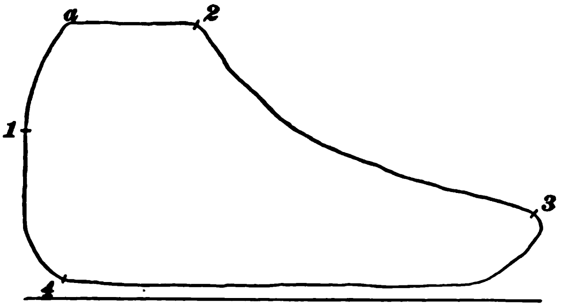


FIG. 273.

1279. Case III.—Cut-off too early. See Fig. 270. Here the steam expands below the back-pressure line and forms a loop. This makes the compression too early also. The figure shown is drawn as if there were no inside lap. With lead, the curve will form a loop, as shown at *A*; without lead, the compression line will extend to *F*. The remedy is to reduce the amount of outside lap. When computing the M. E. P., the areas of both loops must be subtracted from the area *A B C A*.

1280. Case IV.—Cut-off too late. See Fig. 274. Here it will be noticed that the terminal pressure is very high.

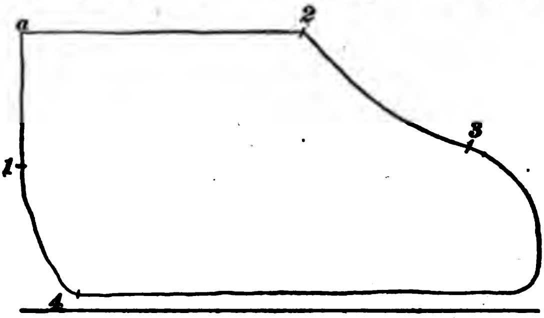


FIG. 274.

When this is the case, a great deal of the benefit of expansion is lost, with its consequent waste of steam.

Rule for Cases III and IV.—Make the cards alike for both ends of the cylinder. For a too early cut-off, lower the boiler pressure or decrease the number of revolutions per minute. For a too late cut-off, raise the boiler pressure or increase the number of revolutions per minute. The cut-off is most correctly equalized by making the terminal pressure at both ends of the cylinder the same.

Case V.—See Fig. 272.

Case VI.—See Fig. 273.

Rule for Cases V and VI.—Arrange the valves so that one-half of the fall of pressure occurs before the piston starts back on the return stroke.

1281. Case VII.—Compression too early.

Fig. 275 shows the effects of too early compression. A loop is formed, as shown in Fig. 270. The area of this loop must be subtracted from the larger area in computing the M. E. P. With the same cut-off and the proper amount of compression, the area gained would be *a b 4 a*, included between the line *4 a* and the dotted line *a b 4*, plus the area of the loop. The remedy in this case is to decrease the amount of inside lap. The required amount of compression depends upon the speed of the engine, slow-running engines not requiring so much compression as the high-speed engines. In any case, the compression should not extend above the initial or boiler pressure.

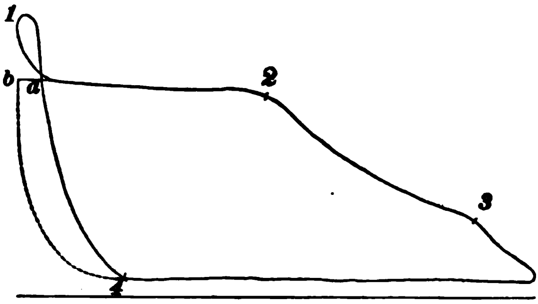


FIG. 275.

It is good practice to compress to about $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{6}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines.

All of the above faults are due to valve setting, and can be detected as soon as the indicator is applied.

1282. With a plain slide-valve it will be found that if one of the events of the stroke is early or late, the others are liable to be so also; for example, an early admission usually produces an early release and compression.

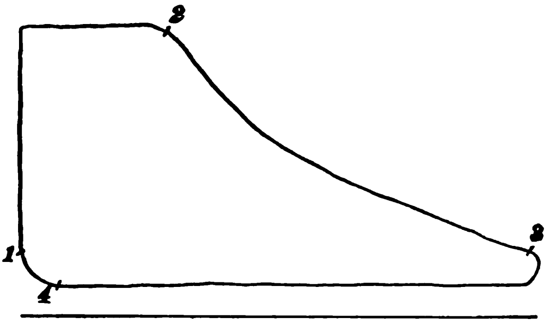


FIG. 276.

When the steam line falls abruptly, as shown in Fig. 272, it may be inferred that the steam is throttled; i. e., either the steam pipe or the port is too small for the required duty. A very high piston speed would also produce this effect.

The card shown in Fig. 276 indicates that the back pressure is excessive. This may be the case when the exhaust port is too small or when the exhaust steam is used for heating purposes, and, in consequence, has to be pushed through coils of pipe.

STEAM CONSUMPTION.

1283. The indicator card also enables us to find approximately the amount of steam consumed by the engine. In referring to the steam consumption, it is customary to take as a unit the *steam consumed per horsepower per hour.*

Take a point *a* on the expansion line before the release (see Fig. 277); measure the pressure from the vacuum line, and from column 6 of the steam table find the weight of a cubic foot at that pressure. The cubic contents of the cylinder (including the clearance) up to the point *a*, multiplied by the weight per cu. ft., must give the weight of steam in the cylinder at this instant. Were it not for compression and cylinder condensation, the above weight would represent the steam consumed per stroke. On account of compression, some steam is saved by the early

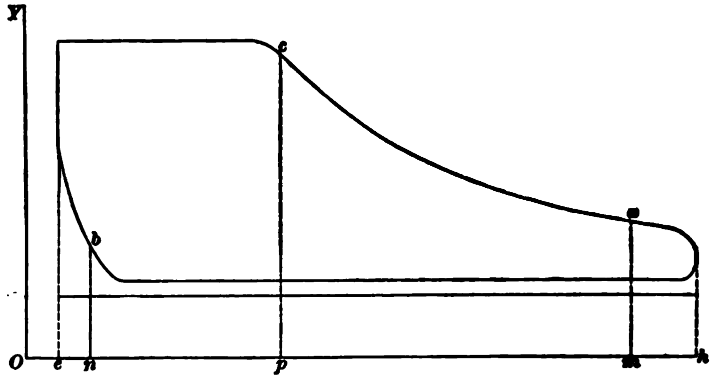


FIG. 277.

closure of the exhaust port. To find its weight, take a point *b* on the compression curve, measure its pressure from vacuum, as before, and compute the weight of the steam in the cylinder up to *b*. Subtract this from the weight first obtained, and the difference will be the weight of steam per stroke, accounted for by the indicator. Multiply this weight per stroke by the number of strokes per hour and divide by the I. H. P. of the engine. The result will be the steam used per I. H. P. per hour.

EXAMPLE.—Fig. 277 represents an indicator diagram taken from an engine with an 18" × 24" cylinder, running at 120 revolutions and developing 130 horsepower. The clearance is 5%. Find the steam consumption per I. H. P. per hour.

SOLUTION.—Project the two ends of the diagram perpendicularly upon the vacuum line, as at *e* and *h*; *e h* is then the length of the diagram. Lay off *e O* equal to the clearance—that is, equal to 5% of *e h*. Draw *O Y* perpendicular to *O h*. Take the point *a*, near the point of release, and measure the distances *a m* and *O m*. Take the point *b*, somewhere on the compression line, and measure the distances *b n* and *O n*. The measurements are found to be:

$$a\,m = 0.71 \text{ inches};$$
$$O\,m = 3.17 \text{ inches};$$
$$b\,n = 0.6 \text{ inches};$$
$$O\,n = \tfrac{1}{3} \text{ inch}.$$

The length of the diagram $= e\,h = 3\tfrac{1}{3}$ inches; the length of the stroke is 2 feet. Hence, each inch of the length of the card equals $\frac{2}{3\frac{1}{3}}$ $= .6$ foot of stroke. The scale of the indicator spring is 45. Hence, the above measurements reduced to pressures in pounds per square inch and feet of stroke become:

$$a\,m = .71 \times 45 = 31.95 \text{ pounds};$$
$$b\,n = .6 \times 45 = 27 \text{ pounds};$$
$$O\,m = 3.17 \times .6 = 1.9 \text{ feet};$$
$$O\,n = \tfrac{1}{3} \times .6 = .2 \text{ foot}.$$

The area of the piston is $18^2 \times .7854 = 254.47$ sq. in. $= \frac{254.47}{144} = 1.767$ sq. ft. Consequently, the volume of steam in the cylinder, when the piston is at the point represented by *a*, is $1.9 \times 1.767 = 3.3573$ cu. ft. The volume, when the piston is at *b*, is $.2 \times 1.767 = .3534$ cu. ft. The weight of a cubic foot of steam at an absolute pressure of 31.95 pounds per sq. in. is found from the steam table to be .078723 pound; and at a pressure of 27 pounds the weight is .067207 pound. Hence, the weight of the steam in the cylinder is $.078723 \times 3.3573 = .264297$ pound; while the weight of steam saved by compression is $.067207 \times .3534 = .023751$ pound. The steam used per stroke is, therefore, $.264297 - .023751 = .240546$ pound, and the amount used per I. H. P. per hour is

$$\frac{.240546 \times 120 \times 2 \times 60}{130} = 26.645 \text{ pounds.}$$

Suppose the weight of the steam in the cylinder to be calculated by taking the point c, near the point of cut-off. $c p = 1.59$ inches $= 1.59 \times 45 = 71.55$ pounds; $O p = 1\frac{1}{3}$ inches $= \frac{4}{3} \times .6 = .8$ foot of stroke. The volume of steam in the cylinder when the piston is at c is, therefore, $.8 \times 1.767 = 1.4136$ cubic feet. One cubic foot of steam at the pressure of 71.55 pounds, absolute, weighs .168009 pound. The weight of the steam in the cylinder is, therefore, $.168009 \times 1.4136 = .237498$ pound. Subtracting the steam saved by compression, the steam used per stroke is $.237498 - .023751 = .213747$ pound, and the steam per I. H. P. per hour is

$$\frac{.213747 \times 120 \times 2 \times 60}{130} = 23.677 \text{ pounds}$$

Now, unless the valve leaks, the weight of the steam when the piston is at a can be no greater than when it is at c, since no fresh steam has been allowed to enter; but the calculation shows that there is .264297 pound in the cylinder when the piston is at a, and only .237498 pound when the piston is at c. This shows that $.264297 - .237498 = .026799$ pound has been condensed to water by the time the piston has arrived at c, but has been re-evaporated before the piston arrives at a. Hence, by calculating the water consumption at cut-off, and then at release, a good idea of the amount of cylinder condensation may be obtained. If the steam used by the engine be actually caught and weighed and then compared with the weight as calculated from release an idea may be obtained of the amount of condensation at release. The computed consumption is always less than the actual consumption.

1284. Where there is a sufficient amount of compression, the work may be simplified by taking the two points a and b at the same height above the vacuum line, as shown in Fig. 278. Since the absolute pressure at a and b is the same, the clearance may be left entirely out of account, and the volume to be used in the computation will be $\frac{l}{L}$ times the volume of the cylinder, or, in other words, $\frac{l}{L} \times$ length of stroke × area of piston. When this method is used, the steam consumption may be found directly from the following formula:

$$Q = \frac{13{,}750\, l\, W}{P L}, \qquad \textbf{(101.)}$$

in which Q is the number of pounds of steam consumed per horsepower per hour, W the weight of a cubic foot of steam at the absolute pressure a, and P the M. E. P.

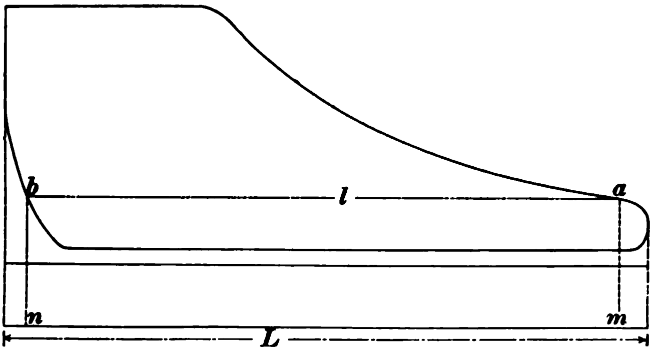


FIG. 278.

EXAMPLE.—From a card taken from an 18½" × 30" engine, the following measurements were obtained (see Fig. 278): $a\,m = .667$ inch; $l = 3.08$ inches; $L = 3.5$ inches; M. E. P. = 35 pounds. What is the steam consumption per I. H. P. per hour?

SOLUTION.—The indicator card being taken with a 45 spring, the pressure at a is $45 \times .667 = 30$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .0742 pound. Substituting in formula **101**,

$$Q = \frac{13{,}750\, l\, W}{P\, L} = \frac{13{,}750 \times 3.08 \times .0742}{35 \times 3.5} = 25.65 \text{ lb. Ans.}$$

EXAMPLES FOR PRACTICE.

1. Size of engine, 12" × 20"; length of card L, 3.4"; length l, 2¼"; height $a\,m$, ⅝"; R. P. M., 230; spring, 30; M. E. P., 18 lb. per sq. in. What is the steam consumption per I. H. P. per hour?

Ans. 25.63 lb. per I. H. P. per hour.

2. Size of engine, 12" × 12"; M. E. P., 51.1; length of card L, 2.6"; length l, 1.8"; height $a\,m$, .7"; R. P. M., 350; spring, 70. What is the steam consumption per I. H. P. per hour?

Ans. 21.92 lb. per I. H. P. per hour.

2. If, in the above engine, example 2, the pressure at cut-off is 110 lb., absolute; the clearance is 5%; the length of the diagram to the point of cut-off is .7"; the pressure at a point on the compression curve is 40 lb., absolute, and the distance of this point from the end of the diagram is .14", what is the steam consumption per I. H. P. per hour at cut-off? Ans. 19.44 lb. per I. H. P. per hour.

THE WHEELOCK ENGINE.

1285. From what has been previously stated, it should now be apparent that the clearance should be as small as may be consistent with construction, and that the valve port opening should be as large and the ports themselves as short as possible. All these conditions are fulfilled in Corliss engines. Another engine which fulfils these conditions, and has fewer parts in connection with its valve gearing, is the Wheelock engine. As this engine is much used in rolling mills and in other places where very heavy duty is required, and is besides a fine example of modern engine practice, a brief description of its valve gearing will now be given.

1286. Fig. 279 shows a side view of the engine complete without the fly-wheel. In the following description, *A* is the eccentric; *B* the eccentric rod; *C* the governor; *E* a lever to which the eccentric rod is attached and which alters the throw of the eccentric as in the case of Fig. 257; *D* the rod which connects this lever with the rockers *G* and *G'*, which move the valves; *H* and *H'* the crab claws; *I* and *I'* the dash-pots; *J* the pilot valve (or valve which admits and shuts off steam from the steam chest. This term is used when a second valve, called a throttle valve, is situated in the engine room, and which is used to shut off the steam from the boiler); *K* the governor rod; *M M* the floor, and *L L'* the exhaust pipes—two in this case.

In Fig. 280, a section through the valve port is given, also a top view of the valve itself. This section is taken at the back end of the cylinder, and shows the steam valve *a* shut, and the exhaust valve *b* open. The piston is, consequently, moving to the right, and the steam is flowing into the exhaust pipe *L'*. The shape of the valves is shown

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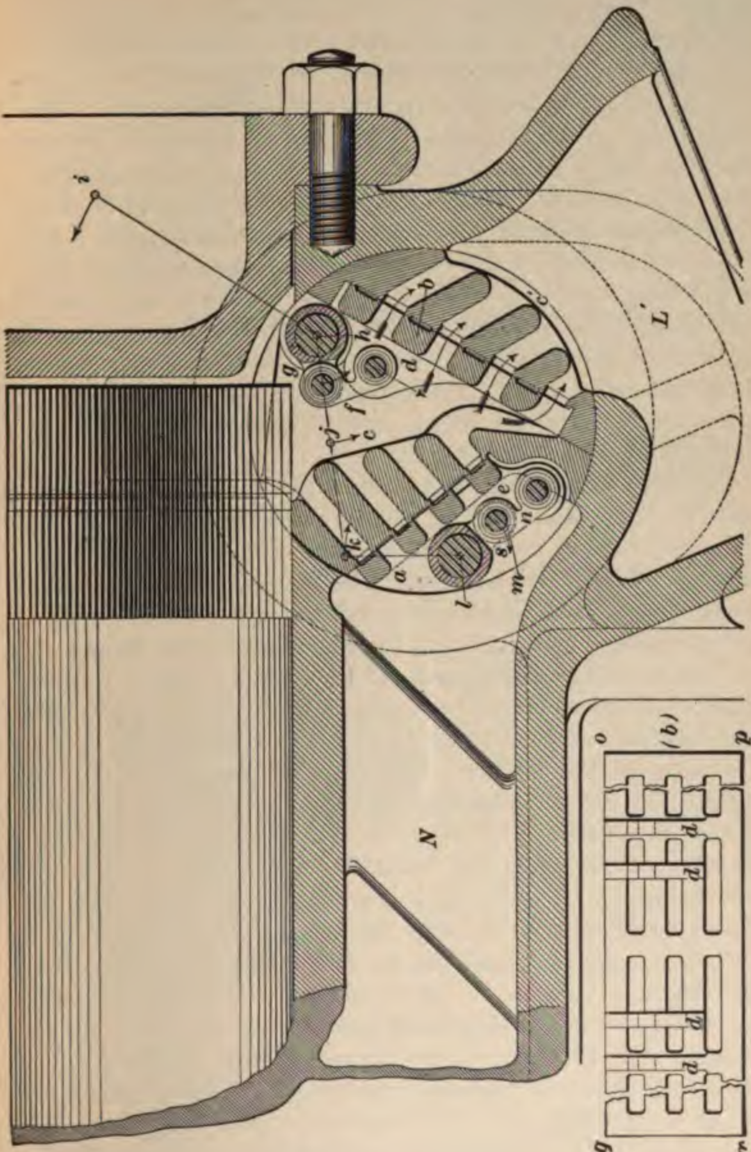

FIG. 280.

at (*b*). They are flat plates. The distance *o p* is the *length* of the valve, and *o q* the *breadth*. The breadth, as will be seen, is several times the length, and persons unfamiliar with this subject would naturally call *o q* the length. The length of a valve, however, is always measured in the direction in which it moves; hence, *o p* is the length. Steam is admitted to the cylinder and exhausted from it through the holes in the valves. The form of those parts of the valve marked *d* is clearly shown in the sectional view. Two links, *f* and *n*, are fitted between them on each valve, and secured by passing a round pin through the lugs and links. In the sectional view, *c* and *c'* are castings which form the valve seats. The ends of the valves *o p* and *q r* are fitted into grooves which compel any movement of the valves to be in a straight line.

Fig. 281 shows in greater detail the rocker *G'* and crab claw *H'*, Fig. 279. In Fig. 280 only the center lines *i h*, *h j*, *j k*, and *k l* have been given, in order to prevent confusion through multiplicity of lines. The pins, whose centers are *h* and *l*, are fixed in position, and the other three centers are movable. The pins *h* and *l* pass through the flange *P*, and to them are rigidly attached the links *g* and *s* respectively. Suppose the end *i* of the rocker is caused to move in the direction of the arrow; *j* and *k* will also move in the direction of the arrows. The pin *h*, being keyed to the rocker, will rotate in its bearing and cause the small crank *g* (see Fig. 280), which is keyed to *h*, to move downwards. This forces the link *f* and valve *b* downwards. This same movement of *G'* forces *Q'* to the right (see Fig. 281), as shown by the arrow, and thus rotates the pin *l*. This rotation of *l* causes the small crank *s* (Fig. 280) to move to the left, and the valve *a* to move upwards. Both valves continue to move in the directions shown until the point *i* has reached the extreme of its travel to the left, which is about 60° from its present extreme right-hand position. When *G'* starts on its return movement, the valves *a* and *b* also start back. Movement is imparted to *G* and *G'* by means of the eccentric *A* and the connections *B*, *E*, and D.

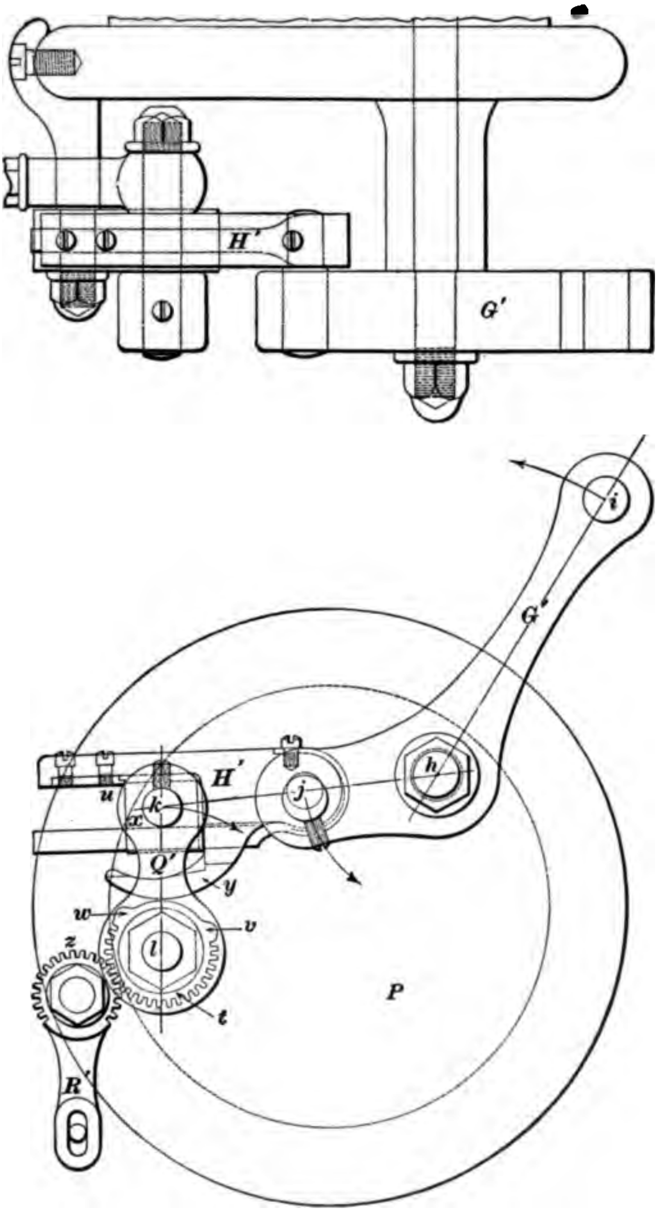

FIG. 281.

To understand the action of the governor in varying the cut-off, consider Figs. 279 and 281. In Fig. 281, the arm R' is seen to have teeth on the circumference of its upper end, which engage with another gear, t. This last gear turns loosely on the pin l, and has two projections, v and w, called *toes*. Fitting in crab claw H' is a block x, which is kept in position by the upper edge u of the crab claw. This block is connected to the dash-pot I' by means of the rod S'. Suppose that part of the load on the engine is taken off; the speed will increase, the centrifugal force of the revolving governor balls will also increase, and, in so doing, the balls fly outwards; raise a spindle, to which they are connected by means of the lower links, and cause the arm F to move to the left by means of a series of levers not shown in the figure. When F moves to the left, it carries with it the rod K, and causes R and R' to move to the left also. This movement causes the gear z to revolve to the right and the loose gear t to revolve to the left. The curved arm y of the crab claw being in a lower position than shown, owing to a movement of G' to the right, the toe v strikes this arm, raises the crab claw and disengages the block x, which is immediately pulled to the left by the dash-pot, and with it the arm Q', which actuates the valve a. The valve is then thrown to the extreme position shown in Fig. 280, and the steam is cut off from the cylinder.

When the speed of the engine is uniform, the valve cuts off in the same manner as an ordinary slide-valve, by reason of its having lap and lead. The governor varies the cut-off only when the speed of the engine changes.

CORLISS VALVE GEAR.*

1287. The **Corliss valve gear** is used in a large number of engines. In Fig. 282 is shown a side elevation of this valve gear, and in Fig. 283 a section through the cylinder and valves. It has four separate and distinct valves. Two of these, v and v', Fig. 283, connect directly with the steam chest d and steam pipe s, and are called steam valves. They

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are rigidly connected with the cranks *N* and *N'* (Fig. 282), *N'* being removed in order to show more clearly the disengaging link *I'*. The other two valves, *r* and *r'*, Fig. 283, connect directly with the exhaust chest *l* and the exhaust pipe *o*, and are called exhaust valves; they are rigidly connected with the cranks *M* and *M'*, Fig. 282. All the valves are cylindrical in form, and extend across the cylinder above and below, respectively.

A, Fig. 282, is a disk or wrist-plate which is made to rock upon a stud *C*, by the eccentric rod *B*, connecting it with an eccentric on the crank-shaft.

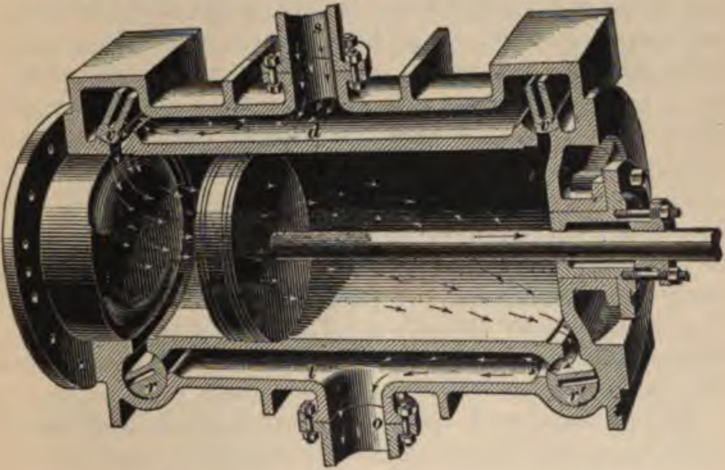


FIG. 283.

There are four valve stems: *E* and *E'*, which connect the wrist-plate *A* with the bell cranks *H* and *H'* of the steam valves, and *F* and *F'* which connect the wrist-plate *A* with the cranks *M* and *M'* of the exhaust valves. The valve stems can be lengthened or shortened as the case may require, and the action of any one valve regulated independently of the other three. As the wrist-plate *A* rocks backwards and forwards, the exhaust valves *R* and *R'*, which are rigidly connected with their cranks *M* and *M'*, rock with it. The bell cranks *H* and *H'*, which are provided with the disengaging links shown at *I* and *I'*, are also given this rocking motion, and by hooking on to the blocks *B* and *B'*, which are rigidly connected to the cranks *N* and *N'*, open the steam valves *V* and *V'*.

The projections *a* and *a'*, on the two trip collars *G* and *G'*, unhook these disengaging links *I* and *I'*, after they have rotated the valves *V* and *V'* through a certain angle, and the cranks *N* and *N'* are pulled back to their first positions by the vacuum air dash-pots *P* and *P'*, against the resistance of which the valve cranks *N* and *N'* were raised. The movements of the valves open and close the steam and exhaust ports of the cylinder at the proper intervals. The pins of the valve stems are so located on the wrist-plate that the steam valves *V* and *V'* have their quickest movement while the exhaust valves *R* and *R'* have their slowest movement, and the exhaust valves have their quickest movement while the steam valves have their slowest movement. As a consequence of this arrangement, the steam and exhaust valves have entirely independent movements, and the inlet ports may be suddenly opened full width by the quick movement of the steam valves, while the exhaust valves are practically motionless. The advantage of this valve gear is that it permits an earlier cut-off, with a greater range, a more perfect steam distribution, and a smaller clearance space than is attained with the plain slide-valve.

Engines fitted with the Corliss valve gear cannot run at much more than 90 revolutions per minute.

GOVERNORS.

1288. When a steam engine is running at a uniform speed, the work done by the steam in the cylinder must just equal the resistance overcome at the fly-wheel rim. Should the resistance become less than the work, the amount of work in excess of that necessary to overcome the resistance would cause the moving parts to move faster and faster, and the engine would "race" or "run away." If, on the contrary, the resistance should exceed the work, the engine would slow down, and finally stop. The work required of the engine can not, of course, remain always constant; hence, it is necessary to have some means of automatically adjusting the steam supply to the variation of the resistance. This is accomplished by the *governor.*

Steam engine governors may be divided into two classes: (1) **throttling governors,** which throttle the steam in the supply pipe, and (2) **automatic** or **adjustable cut-off governors,** which regulate the steam supply by changing the point of cut-off of the valve.

1289. The ordinary throttling governor, shown in Fig. 242, consists of a balanced throttle valve placed on the steam pipe *s*; this valve is attached to the spindle *k*, at the upper end of which are the two **fly-balls** *m*, *m*, the spindle and fly-balls forming what is known as a **revolving pendulum.** The spindle and balls are driven from the main shaft by the belt *p* and the bevel wheels *o*. If the engine moves faster than the desired speed, the fly-balls are forced to revolve at a higher speed, and will, consequently, move outwards and upwards through the action of centrifugal force. This forces the spindle *k* downwards, and partly closes the throttle valve. The engine thus takes less steam, and the speed falls to the desired point, the governor balls in the meantime returning to their original position. Should the resistance become greater than the power of the engine, it slows up slightly, the balls drop and open the valve wider. More steam is admitted, and the engine immediately regains its original speed. The chief objection to the throttling governor is that the steam is wire drawn. The term **wire drawn** is applied to any case in which the steam pressure is reduced owing to the insufficiency of valve opening. The term **throttled** is also applied to such cases. Steam is more or less wire drawn in all engines fitted with plain slide valves—the rounded corners on the diagram prove this; this is because the movement of the valve is comparatively slow when closing the ports. With Corliss and other releasing gear engines, the valve movement at cut-off and release is very rapid, and the wire drawing very slight.

1290. In throttling engines (plain slide-valve engines), the steam line of the indicator diagram always inclines towards the atmospheric line instead of being nearly parallel with it, as in the cards previously given. In consequence of this, the mean effective pressure is less. Another form of a governor is shown in Fig. 284. This is the well-known Pickering governor. There are three balls, and they move outwards against the resistance of gravity and the three flat springs *s*. In so doing they lower the valves *v* and *v'*. The steam enters at *K*, flows in the direction of the arrows, and then through *J* into the steam chest. Since steam is on both sides of the valves, they are *balanced.* The object of using two valves instead of one is to afford a large opening with a small lift of the valve.

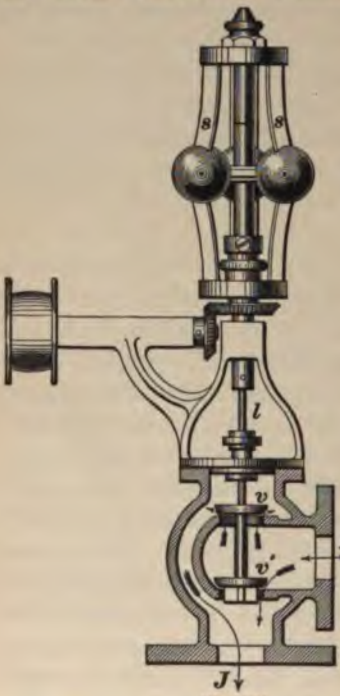


FIG. 284.

1291. As applied to the Corliss type of engines, the revolving pendulum or fly-balls vary the point of cut-off instead of throttling the steam supply. The method of operation is shown in Fig. 285; the fly-balls *m*, *m* are here given a rotary motion by a belt and gear wheels, in the same manner as in Fig. 242.

Let it be supposed that the engine is running at its proper speed. The fly-balls will then be held in their normal position by the balance existing between the centrifugal and gravity forces acting on *m* and *m*. Suppose, now, the speed of the engine increases from any cause whatever, the centrifugal force acting on the fly-balls will also increase and will continue to pull them out, that is, to increase the diameter of the circle in which they rotate, until a new balance is effected between it and the attraction of gravity. This movement of the fly-balls will be transmitted to the lever *D*, causing it to turn slightly about its center in the direction of the arrow *X*. The movement of *D* will cause the trip collars *G* and *G'* (see Fig. 282) to turn through a small angle in such a direction that their projections *a* and *a'* will unhook the disengaging links *I* and *I'* earlier in the stroke. This will cause the point of cut-off to occur earlier in the stroke, and a decrease in the speed of the engine, on account of the reduction in the amount of steam admitted to the cylinder, and an increased ratio of expansion of the steam under the same initial pressure. Should the speed from any cause diminish, a reverse operation would be the result. The fly-balls would drop slightly; *D* would turn as indicated by the arrow *U*, and the trip collars *G* and *G'* would be rotated in such a manner as to cause their projections *a* and *a'* to unhook the disengaging links *I* and *I'* later in the stroke; the cut-off would then occur later in the stroke, and a diminished ratio of expansion at the same pressure would again bring the speed up to its proper point.

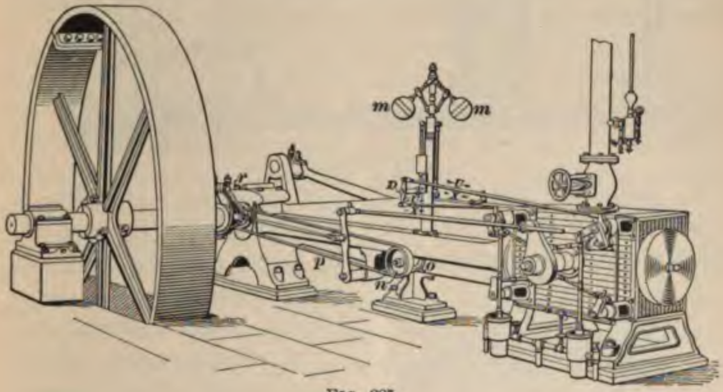


FIG. 285.

1292. All engines which require a disengagement of some of their parts in order to affect the cut-off by the action of the governor, as the Wheelock, Corliss, etc., are termed **releasing gear** engines, in contradistinction to those in which the cut-off is affected directly by the governor, without the intervention of dash-pots, cams, weights, etc. The valve gearing of this latter class is termed **positive.** Engines of the releasing gear type are limited to a speed of 90 revolutions per minute, or less, since, if the speed is increased beyond this limit, the valves will not work properly. Hence, to obtain a high piston speed, it is necessary to make the cylinders very long. Engines are sometimes required to have a high rotative speed, as in the case of electric lighting,

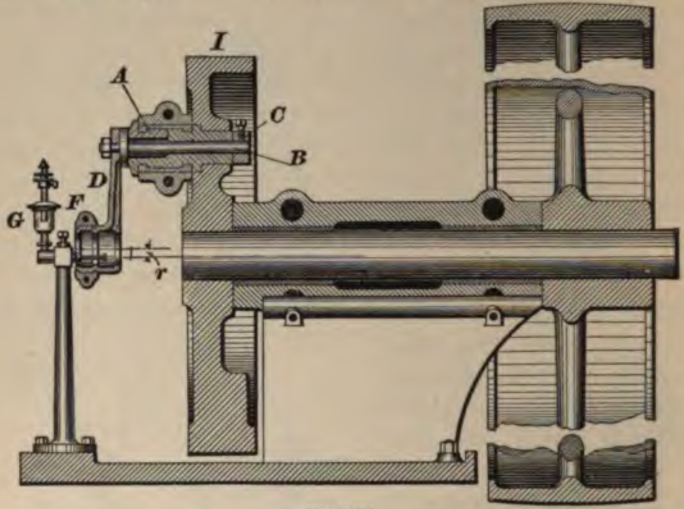


FIG. 286.

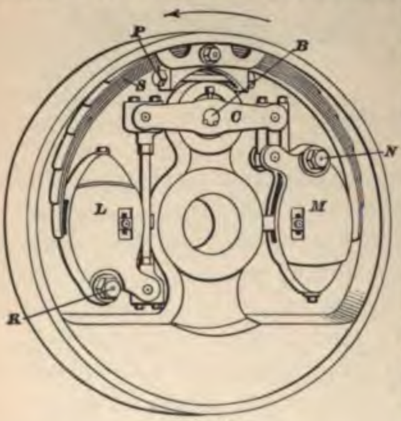


FIG. 287.

and those having positive valve gears respond fully to all demands. By **high rotative speed** is meant a large number of revolutions per minute. The rotative speed of a positive geared engine may be 400 revolutions per minute, or more, and still be sensible to variations in the load.

In automatic cut-off engines having positive valve gears, the governors are of the type known as **shaft governors.** There are many varieties of the shaft governor, but only one will be described here.

Fig. 286 shows a section through the shaft and fly-wheels, and Fig. 287 a perspective view of the fly-wheel to which the governor is attached. Engines of this class usually have two fly-wheels—one termed the *governor wheel* and the other the *belt* or *band wheel.* If necessary, both wheels can drive belts, but it is not usual to use the governor wheel for this purpose, unless it is the only fly-wheel. By reference to Fig. 286, it will be seen that the crank pin *A* is hollow, and that a rod *B* passes through it, to which is attached a crank *D*, on the left end. This crank has an enlarged end and really forms the eccentric, *F* being the eccentric sheave. The manner in which the enlarged end of *D* forms the eccentric is this : The center line of the enlarged end, and, consequently, of the sheave *F*, does not coincide with the axis of the engine shaft; hence, since the arm *D* is always in the relative position in regard to a point on the governor wheel, unless the governor acts in a manner to be hereafter described, the distance *r* between these two center lines remains the same as the crank and governor wheel revolve. The distance *r* is the eccentric radius, and $2r$ is the throw of the eccentric = the travel of the valve.

In Fig. 287, *C* is a bar keyed to the rod *B* which passes through the crank-pin. On each end of *C* is suspended a rod; both of the rods are attached to the heavy weights *L* and *M*, which are pivoted at *K* and *N*. *S* is a finely tempered spring which presses against the weights. These two weights and rods, the bar *C* and spring *S*, form the governor, which acts in the following manner: As the governor wheel revolves in the direction of the arrow, the centrifugal force generated in the weights *L* and *M* tends to cause them to turn on their pins *K* and *N*, but this motion is resisted to a certain extent by the spring *S*. The spring holds the weights in equilibrium against the centrifugal force when the speed of the engine has reached the desired limit. For a speed beyond this limit, the centrifugal force overcomes the force of the spring and the weights move outwards; but, for a speed below the limit, the force of the spring overcomes the centrifugal force and the weights move inwards. The eccentric is never in the position shown in Fig. 286 except when the engine is not running; it usually occupies a position such that the distance *r* is greater when the engine is running. Suppose the engine is running at its rated speed under its rated load. The weights and the eccentric will occupy a certain position, and the valve will travel back and forth an amount corresponding to twice the distance *r*, whatever that may be. All the operations will be exactly the same as in the case of the ordinary slide valve with a fixed eccentric. Suppose that an extra load is thrown upon the engine. The speed will momentarily slacken; the weights will move inwards; the left end of the bar *C* will turn towards the shaft and the right end away from the shaft. This causes the rod *B* to rotate in its bearing in such a manner that the arm which carries the eccentric is forced further away from the shaft; the distance *r* is increased and, consequently, the valve travel; the cut-off is later; more steam is admitted, and the engine almost instantly regains its former speed, the weights returning to their former position. Should the speed be increased, the reverse takes place. The balls *L* and *M* fly outwards and rotate the lever *C*, rod *B*, and arm *D* in the opposite direction; the distance *r* is lessened; the cut-off takes place earlier, and the engine regains its former speed.

The speed of the engine may be varied by means of the nut *P*. By tightening or loosening *P*, the tension of the spring is altered, and with it the speed of the engine.

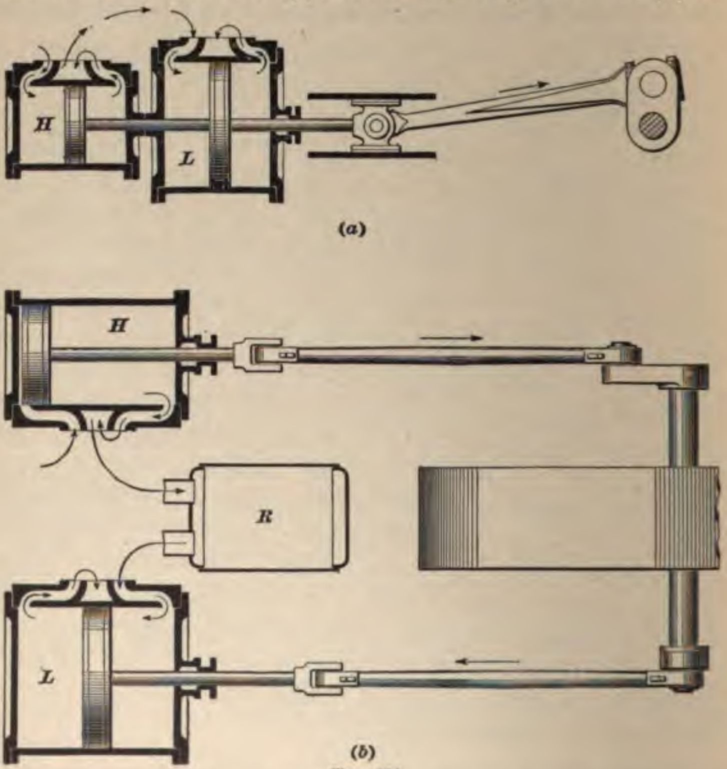
G, Fig. 286, is an oil cup which is supported by the stand E. The oil drops into the hollow eccentric and flows down the arm D, when it is revolving, into the bearing, lubricating the rod B so that the governor may work freely.

COMPOUND ENGINES.

1293. The thermal efficiency of the steam engine has been shown to be $\frac{T_1 - T_2}{T_1}$. T_2 can not, in practice, be lower than the absolute temperature of the condenser. Hence, the only way of increasing the fraction $\frac{T_1 - T_2}{T_1}$ is to increase T_1, or, in other words, to increase the temperature and, consequently, the pressure of the entering steam. Following out this idea, steam pressures have steadily increased from 8 to 10 pounds, in the time of Watt, to 150 to 200 pounds per square inch, the pressures used in modern locomotives and marine engines. But here the evil of cylinder condensation again appears; for, by increasing the range of temperature, $T_1 - T_2$, the loss by cylinder condensation is largely increased. To see this clearly, let the pressure of the steam passing into the condenser be 4 pounds above vacuum; its temperature is about 153°. Let the pressure of the entering steam be say 60 pounds above vacuum; its temperature is about 293°. The fall in temperature is $293° - 153° = 140°$, nearly. Suppose, however, the entering steam has a pressure of 200 pounds above vacuum; its temperature would then be 382°, nearly, and the fall in temperature during the stroke would be $382° - 153° = 229°$. Now, it is plain that a great deal more of the incoming steam must condense to raise the temperature of the cylinder walls back from 153° to 382°, than to raise them from 153° to 293°. *Hence, increasing the range of temperature increases the loss due to cylinder condensation.*

1294. To obtain the advantages of a high pressure, and, at the same time, avoid the loss due to cylinder condensation as much as possible, the steam may be allowed to expand successively in two or more cylinders. The fall of temperature is thus divided between the two or more cylinders and, consequently, the loss from condensation in both, or all of them, is made considerably less than it would be if the same fall of temperature was allowed to take place in one cylinder. When the expansion takes place in two cylinders, the engine is said to be **compound**; if the expansion takes place in three cylinders, the engine is said to be **triple expansion,** and if in four cylinders, **quadruple expansion.**

1295. Compound engines are usually made in one or the other of the two types shown in Fig. 288. In (*a*) the two cylinders are placed in line, the two pistons being attached to the same piston rod. *H* is the cylinder which first



(*a*)

(*b*)

FIG. 288.

receives steam from the boiler; it is called the **high-pressure** cylinder. After the steam has expanded in *H*, it passes to the larger cylinder *L*, which is called the **low-pressure** cylinder; from here the steam is exhausted into the atmosphere or into a condenser.

Fig. 288 (*b*) shows what is known as the **receiver** compound engine. The steam enters the high-pressure cylinder *H* from the boiler; exhausts into a separate vessel *R*, called the **receiver;** from there it passes to the low-pressure cylinder *L*, and finally exhausts into the atmosphere or into a condenser.

A receiver compound engine has two piston rods and two cranks; the cranks may be placed at any angle with each other. The compound engine, without a receiver, may have one piston rod and crank, as shown in the tandem type, or it may have two piston rods and two cranks, the cylinders being placed side by side. In any compound engine, without a receiver, the two pistons must begin and end their stroke at the same time, and the two cranks must be together, or placed 180° apart.

1296. When one cylinder is placed behind the other, as shown in Fig. 288 (*a*), the engine is called a **tandem compound.** When the cylinders are placed side by side, as shown in (*b*), and the piston rods are attached to separate cross-heads, the engine is called a **cross compound;** if both piston rods are attached to the same cross-head, the engine is called a **twin compound.** If any of these types of engines have a condenser, they are called **tandem, cross,** or **twin, compound condensing engines.** Without a condenser, they are called **non-condensing engines.** They all may or may not have a receiver.

1297. In giving the size of a multiple-expansion engine, the stroke is always written last. Thus, a compound engine whose high-pressure cylinder was 11″ in diameter, low-pressure cylinder 20″ in diameter, and stroke 15″ would be expressed as a 11″ and 20″ × 15″ compound. In the same manner a 14″, 22″, and 34″ × 18″ triple-expansion engine would indicate that the diameters of the cylinders were 14″, 22″, and 34″, and that they had a common stroke of 18″.

1298. The **ratio of expansion** of a compound or triple-expansion engine is the ratio between the volume of steam exhausted into the atmosphere, or into the condenser, per stroke, and the volume of steam in the high-pressure cylinder at the point of cut-off.

Let e = ratio of expansion in high-pressure cylinder;
" E = total ratio of expansion;
" v = volume of cylinder receiving steam from the boiler;
" V = volume of cylinder exhausting into atmosphere, or condenser.

Then, $E = \frac{e V}{v}$; **(102.)**

that is, the **total ratio of expansion,** or, as it is usually expressed, the **number of expansions,** is equal to the ratio of expansion of the small cylinder multiplied by the ratio between the volumes of the two cylinders. The total ratio of expansion in a compound engine depends only upon the relative volumes of the cylinders, and the point of cut-off in the high-pressure cylinder; it does *not* depend at all upon the point of cut-off in the low-pressure cylinder. The number of expansions in a compound engine varies from 6 to 12; in a triple-expansion, from 10 to 25.

EXAMPLE.—It is desired to have a total ratio of expansion of 9; the number of expansions in the high-pressure cylinder is 2.72; the volume of the high-pressure cylinder is 6 cu. ft. What must be the volume of the low-pressure cylinder?

SOLUTION.—$E = 9$; $e = 2.72$; $v = 6$.

Substituting in formula **102,** $9 = \frac{2.72\,V}{6}$, or $V = \frac{6 \times 9}{2.72} = 20$ cu. ft., nearly. Ans.

EXAMPLE.—The low-pressure cylinder is four times as large as the high-pressure cylinder, and the real cut-off of the latter is ⅖. What is the total ratio of expansion?

SOLUTION.— $e = \frac{1}{k} = \frac{1}{\frac{2}{5}} = 2.5.$

$$E = \frac{e V}{v} = \frac{2.5 \times 4}{1} = 10. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A compound engine has cylinders 15″ and 25″ diameter by 20″ stroke; that is, its size is 15″ and 25″ × 20″. The clearance in the high-pressure cylinder is 14%, and the apparent cut-off is ⅓. What is the number of expansions? Ans. 6.69.

2. A 28″, 48″, and 74″ × 60″ triple-expansion engine cuts off at ⅖ stroke in the high-pressure cylinder. Clearance in the high-pressure cylinder, 2%. Find the number of expansions. Ans. 17, nearly.

3. If the 26″, 40″, 60″, and 70″ × 72″ quadruple-expansion engine at the World's Fair cut off the steam in the high-pressure cylinder at ¼ stroke, and the clearance in that cylinder was 8%, what was the total number of expansions? Ans. 26⅜, nearly.

COMPOUND ENGINE DIAGRAMS.

1299. In Fig. 289 are shown the ideal diagrams of a tandem compound engine, neglecting clearance and compression. *A B C D* is the diagram from the high-pressure cylinder. Steam enters at the boiler pressure *O A*; at *B*, cut-off occurs, and the steam expands along the line *B C* to the end of the stroke. The steam and expansion lines *A B* and *B C* are precisely like those of a simple engine. At *C*, the exhaust opens. Instead, however, of exhausting into the atmosphere or a condenser, the steam exhausts into the low-pressure cylinder. The low-pressure cylinder is always larger than the high-pressure cylinder; consequently, the volume of steam exhausting from the high-pressure into the low-pressure cylinder is constantly increasing. This is shown in Fig. 290, when the two pistons are at the end of the stroke, as shown at (*a*), the

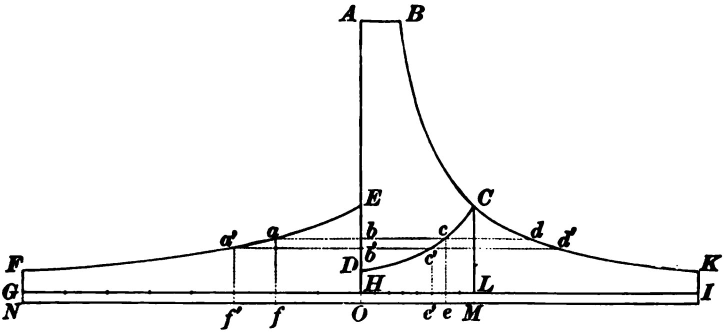


FIG. 289.

steam simply fills the small cylinder *H*; but at the middle of the stroke, as shown at (*b*), the steam fills half of (*a*) and also half of (*b*). Hence, its volume grows greater as the two pistons move to the right.

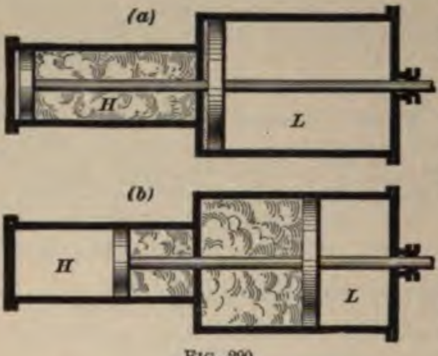


FIG. 290.

1300. Returning now to Fig. 289, the steam, when the piston is at the end of the stroke *c*, just fills the small cylinder; on the return stroke, however, as has just been shown, the volume increases, and hence the pressure falls, as shown by the back pressure line *C D*. At the end *D* of the return stroke, steam again enters the right side of the cylinder *H* from the boiler and raises the pressure to *A*, thus completing the cycle of operation.

C D H L represents the card from the low-pressure cylinder *L*. Since the high-pressure cylinder exhausts directly into the low-pressure cylinder, the back pressure of the former must be the same as the forward pressure of the latter. Hence, *C D* is both the back pressure line of *H* and the expansion line of *L*. At the end of the stroke, the pressure drops from *D* to *H*, *O H* being the pressure in the condenser. The remainder of the diagram is the same as that of a simple engine. In the above description, the common length *O M* of the two diagrams has been taken proportional to the length of stroke.

Suppose now that the length of each diagram be taken to represent the volume of the cylinder to which it belongs. Let *O M* represent the volume of the high-pressure cylinder. Suppose that the volume of the low-pressure cylinder is say 3 times the volume of the high-pressure cylinder. The length of the low-pressure diagram must then be 3 times that of the high-pressure diagram. From *O* lay off *O N* =

$3 \times OM$, and project the expansion line CD into the line EF. This is done by dividing the line OM into a convenient number of equal parts, in this case 8, and ON into the same number of equal parts. At the points of division e, e', etc., erect ordinates cutting CD in c, c', etc. At the points of division O, N erect ordinates fa, $f'a'$, etc. Through c, c', etc., draw lines parallel to MN, cutting af in a, $a'f'$ in a', etc.; a, a', etc., will be points on the required line EF. E and F are, of course, opposite C and D. $EFGH$ represents the low-pressure card to the same scale of pressures and volumes that $ABCD$ represents the high-pressure card—$EFGH$ has been laid off to the left of OA simply for convenience.

1301. The two diagrams may be combined into one in the following manner: Draw a horizontal line, as ad, intersecting both diagrams. The volume of steam in the high-pressure cylinder at the pressure $ec = Ob$ is represented by the length bc; the volume of steam in the low-pressure cylinder at the same pressure is represented by ab. Hence, the total volume of steam at the pressure in question is $ab + bc = ac$. From c, lay off $cd = ab$; then, $ab + bc = bc + cd = bd =$ volume of steam at pressure Ob. In the same manner find that $b'd'$ equals the volume of steam in both cylinders when the pressure is represented by Ob'. By finding a sufficient number of these points, d, d', etc., the curve CK may be drawn. This curve represents the relation between the common pressure in the two cylinders and the total volume of steam in both cylinders, and it will be found that it is simply a continuation of the expansion curve BC of the high-pressure cylinder. It is seen that the combination of these two diagrams forms one large diagram, $ABCKIHA$, equal in area to the sum of the areas $ABCD$ and $EFGH$; for the mean ordinates of $EFGH$, $CDHL$, and $CKIL$ are equal from the nature of the construction; hence, representing the mean ordinate by h, the area of $EFGH = h \times GH$; of $CDHL = h \times HL$, and of $CKIL = h \times LI$. But $GH = 3\,HL$ and $LI = 2$

$H L$; consequently, area $E F G H = h \times 3 H L = h \times H L + h \times 2 H L =$ area $C H I K$. The length of the large diagram represents the volume of the low-pressure cylinder, while the initial pressure $O A$ is the boiler pressure of the steam entering the high-pressure cylinder.

The ratio of expansion of the compound engine of Fig. 289 is $\frac{H I}{A B}$. If the large diagram $A B K I H A$ be considered as the diagram of a single engine, the ratio of expansion is also $\frac{H I}{A B}$. A single engine giving the large combined diagram $A B K I H A$ will do the same work as the compound engine giving the two diagrams $A B C D$ and $E F G H$; but, in order that a single engine may give the diagram $A B K I H A$, the volume of its cylinder must be represented by $H I$—that is, it must be equal to the volume of the low-pressure cylinder of the compound engine, and, further, it must work with an initial pressure $O A$ equal to the boiler pressure of the compound.

The horsepower of a compound engine is approximately equal to the horsepower of a single engine having a cylinder equal in volume to low-pressure cylinder of the compound, and working with the same ratio of expansion and with the same boiler pressure.

In general, it is customary to calculate the horsepower of a compound, triple, or quadruple-expansion engine as if the total expansion took place in the cylinder exhausting into the condenser. This will give a rough approximation to the true horsepower, which will usually be less than the calculated value.

EXAMPLE.—The low pressure cylinder of a compound engine is 30″ × 40″. The boiler pressure is 100 pounds (gauge) and the number of expansions 8. Find the approximate horsepower, assuming the number of revolutions per minute to be 60.

SOLUTION.—Absolute pressure $P = 100 + 14.7 = 114.7$ lb.; take back pressure p, equal to 3 lb. for condensing engine; ratio of expansion $r = 8$.

Substituting in formula 99,

$$\text{M. E. P.} = \frac{.9\,P(1 + 2.3 \log r)}{r} - .9\,p =$$

$$\frac{.9 \times 114.7\,(1 + 2.3 \log 8)}{8} - 9 \times 3 = 37 \text{ pounds, nearly.}$$

Now, using formula 98,

$$\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{37 \times 40 \times 30^2 \times .7854 \times 60 \times 2}{33{,}000 \times 12.} = 317.016 \text{ H.P. Ans.}$$

1302. Fig. 291 shows the ideal diagram of a tandem compound engine, taking clearance and compression into account. The steam and expansion lines $A\,B$ and $B\,C$ of the high-pressure cylinder are similar to those of a single engine. At C the pressure drops slightly as the steam is admitted to the low-pressure cylinder. The back-pressure line of the high-pressure card and the expansion line of the low-pressure card are parallel, the slight difference between them being due to the resistance of the pipe connecting the cylinders. At E the steam is cut off from the large cylinder,

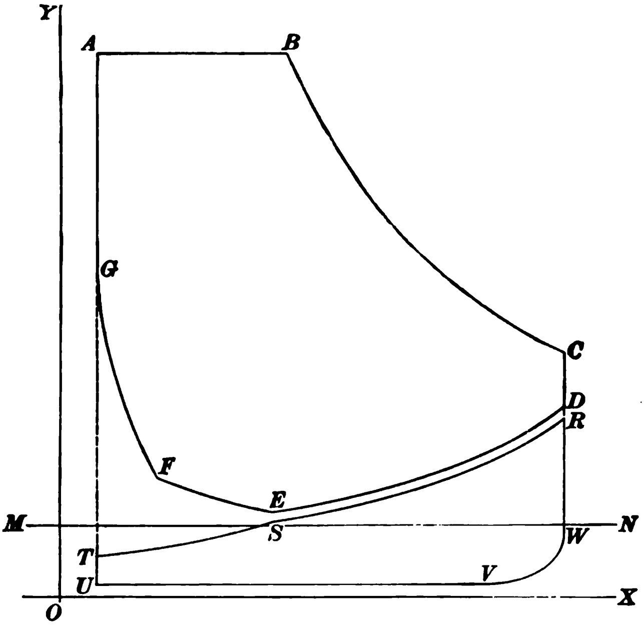


FIG. 291.

and is compressed in the small cylinder, and in the pipe connecting the two. At *F* the exhaust closes, and the steam is compressed in the small cylinder alone from *F* to *G*; at *G* fresh steam again enters.

When the cut-off of the low-pressure cylinder occurs as at *S*, the steam already in the cylinder expands, following the ordinary equilateral hyperbola *S T*. At *T* release takes and the pressure falls to the pressure of the condenser. mainder of the card is the same as for a simple engine.

1303. In Fig. 292 is shown a diagram taken from an actual engine. The similarity between it and the theoretical diagram of Fig. 291 will readily be seen.

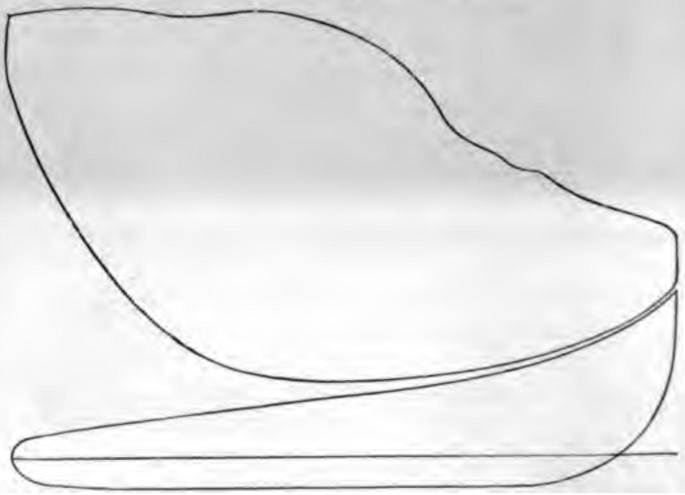


FIG. 292.

1304. The action of the steam in a receiver cross-compound engine may perhaps be better shown by assuming certain conditions to be fulfilled, and from them working out the theoretical diagram.

Suppose that the volume of the low-pressure cylinder is 12 cu. ft.; that the volume of the high-pressure cylinder is 4 cu. ft., and that the volume of the receiver is also 4 cu. ft.; that the steam is to be cut off in the high-pressure cylinder at $\frac{1}{3}$ stroke, and in the low pressure cylinder at $\frac{1}{2}$ stroke; that the boiler pressure is 100 pounds absolute, and the cranks make an angle of 90° with each other. Neglect clearance and compression. The stroke of the high-pressure cylinder begins at *A* (see Fig. 293), the pressure at *A* being 100 lb.; at *B* cut-off takes place, and the steam expands along the equilateral hyperbola *B C*. The volume of steam at *C* is 3 times that of *B*, and, since $p_1 v_1 = p_2 v_2$, the pressure at *C* is $\frac{100 \times 1}{3} = 33.3$ pounds. The steam is released at *C*, and passes into the receiver, where it mixes with steam at receiver pressure. In order to find the resulting pressure of the mixture of the steam in the high-pressure cylinder, and the steam in the receiver, it will first be necessary to find the pressure of the latter. The total ratio of expansion is found as follows: The volume of the high-pressure cylinder at cut-off is $\frac{4}{3}$ cu. ft.; volume of low-pressure cylinder is 12 cu. ft.; number of expansions $= 12 \div \frac{4}{3} = 9$. If clearance had been taken into account, the above would be modified to a certain extent. It can be proven that the receiver exerts no effect whatever upon the number of expansions or upon the final terminal pressure. This is true, no matter what the size of the receiver may be.

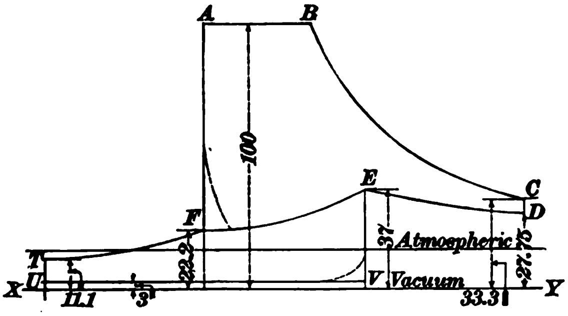


FIG. 293.

Hence, the terminal pressure in the low-pressure cylinder is $\frac{100}{9} = 11.1$ pounds. Since the low-pressure cylinder cuts off at $\frac{1}{2}$ stroke, the pressure at the point of cut-off is

$11.1 \times 1 = p \times \frac{1}{2}$, or $p = 22.2$, the volume of steam at cut-off being half that at end of stroke. Now, just before cutting off, the low-pressure cylinder was receiving steam from the receiver; hence, the pressure in the receiver at the instant of cut-off in the low-pressure cylinder is 22.2 pounds. Since the cranks are at right angles to each other, the high-pressure piston is just at the end of its stroke when cut-off occurs in the low-pressure cylinder. Hence, the steam at 33.3 pounds pressure, on being released from the high-pressure cylinder, rushes into the receiver and mixes with the steam at 22.2 pounds pressure. The pressure of the mixture is found from formula **63,** Art. **1062,**

$$V P = v p + v_1 p_1.$$

The volume of steam in the high-pressure cylinder having a pressure of 33.3 lb. is 4 cu. ft., and that in the receiver having a pressure of 22.2 lb. is 4 cu. ft. The low-pressure cylinder being cut off, the total volume of the mixture is 8 cu. ft.

Substituting, $8 \times P = 4 \times 22.2 + 4 \times 33.3$, or $P = 27.75$ lb. The pressure thus drops from *C* to *D*. The steam is now compressed in the high-pressure cylinder and receiver by the return stroke of the high-pressure piston. The volume of steam at *D* is 8 cu. ft.; when the piston has completed one-half of its return stroke the volume is $4 + 2 = 6$ cu. ft.; hence, the pressure is $27.75 \times \frac{8}{6} = 37$ lb., as shown at *E*. Since at *E* the high-pressure piston is at the middle of its return stroke, the low-pressure piston must be at the beginning of its return stroke. The steam in the receiver and high-pressure cylinder is now admitted to the other end of the low-pressure cylinder, its volume increases as shown in Fig. 290, and its pressure falls accordingly, as shown by the line *E F*, which is a part of the back-pressure line of the high-pressure diagram, and the steam line of the low-pressure diagram. The pressure at *F* has already been found to be 22.2 pounds, and the terminal pressure at *T* to be 11.1 pounds. Hence, the expansion line of the low-pressure diagram is an equilateral hyperbola through *F* and *T*. At *T* the pressure drops to that of the condenser, about 3 pounds; the remainder of the card is the same as for a simple engine.

In Fig. 294 is shown a diagram from a compound marine engine. It will be readily seen that it quite closely resembles the theoretical diagram.

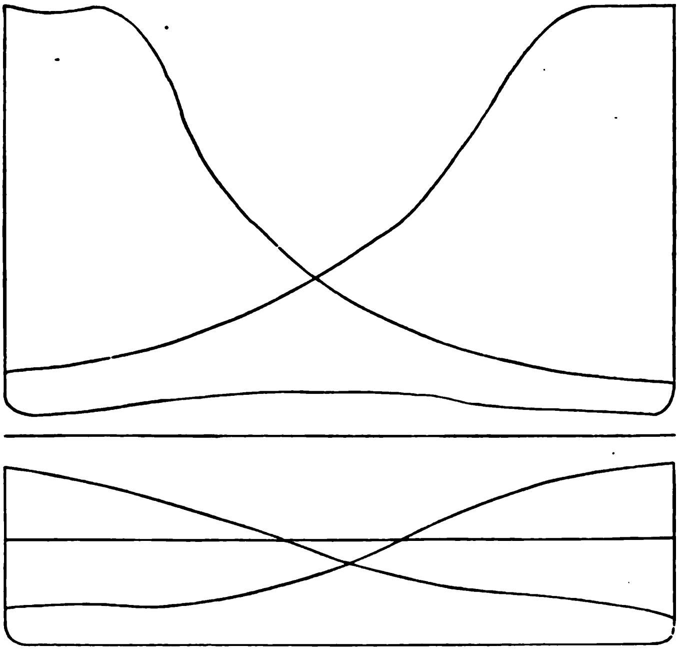


FIG. 294.

USUAL METHOD OF COMBINING THE CARDS.

In order to show the usual method of combining the cards of a multiple (two or more) expansion engine, Figs. 294, 295, and 296 have been drawn. In Fig. 295, *H* is a diagram from the high-pressure cylinder of a Porter-Allen tandem-compound condensing engine, and *L* is a diagram from the low-pressure cylinder. When taking cards from compound or other multiple-cylinder engines, it is usual to use different springs in each indicator on the different cylinders in order to have the height of the cards as nearly alike as possible.

since the larger the diagram, the more clearly can the various points about the diagram be seen. The lengths of all the cards are made the same, if possible. In the present case the lengths of both cards are nearly the same, but a 60 spring was used to take the card *H*, and a 20 spring for the card *L*. Reducing *L* to a 60 spring by dividing the atmospheric line *A B* into 10 equal parts, and making the ordinates through these parts $\frac{20}{60} = \frac{1}{3}$ as long, Fig. 296 is obtained by simply placing the atmospheric line of one card upon that of the other.

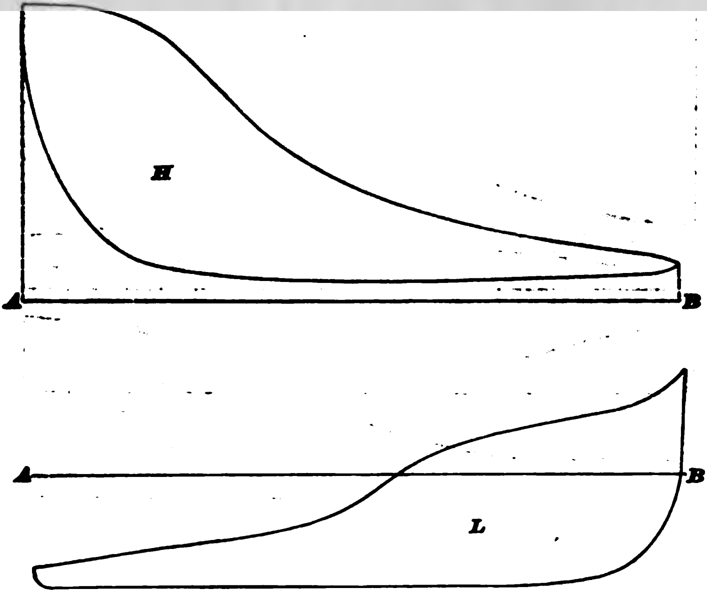


FIG. 295.

1305. In order to study the effects of the expansion, and to determine whether the points of cut-off have been properly located, the two diagrams shown in Fig. 296 must be combined into one diagram. It should be noted that the two diagrams in Fig. 296 are precisely the same as if both had been taken with a 60 spring, and one laid upon the other so that their atmospheric lines coincided. The necessary engine data are as follows: Diameter of high-pressure cylinder, 24 in.; of low-pressure cylinder, 46 in.; stroke, 42 in.; clearance of high-pressure cylinder, 7%; of low-pressure cylinder, 2%. From this it is seen that the volume of the low-pressure cylinder (piston displacement) is 3.66, say $3\frac{2}{3}$ or $\frac{11}{3}$ times that of the high-pressure cylinder. The actual volume of the clearance in both cylinders is about the same, since $.07 \div \frac{11}{3} = .02\frac{1}{3} = 2\%$ of the low-pressure cylinder volume, nearly. In order to prevent the great length of the cards, by increasing the length of the low-pressure card an amount corresponding to the ratio of the volume of the low-pressure to the high-pressure cylinders, it is usual to have

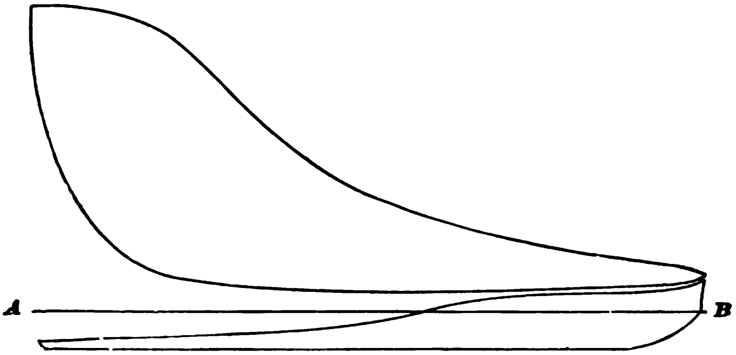


FIG. 296.

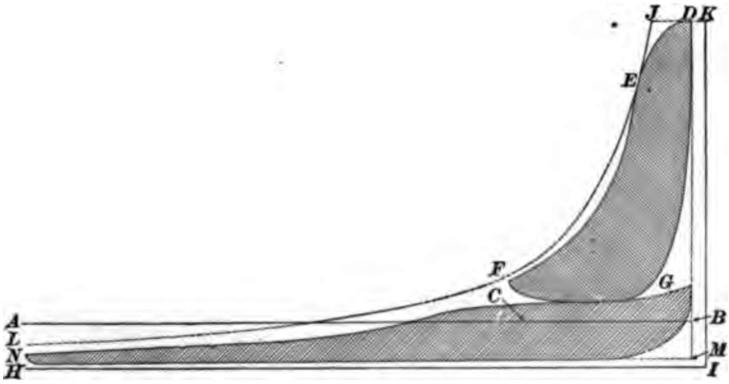


FIG. 297.

the low-pressure card as shown in Fig. 296, and *decrease* the length of the high-pressure card an amount corresponding to the ratio of the cylinder volumes. This has been done in Fig. 297, and in the following manner: Divide *A B* in *H*, Fig. 295, into 10 equal parts and erect ordinates at the points of division. Make *C B*, Fig. 297, equal in length to $\frac{3}{11}$ of *A B*, same figure, and divide *C B* into 10 equal parts. Draw ordinates through the points of division on *C B*, and make them equal in length to the corresponding ordinates on *H*, Fig. 295. Trace a line through the extremities of these ordinates, and the result will be *D E F G D*, or the high-pressure diagram reduced to the volume of the low-pressure cylinder. *E* is the point of cut-off in the high-pressure cylinder, *I K* the clearance line (*D K* being 2% of *A B*), and *H I* is the vacuum line. With *I* as the point of no pressure and no volume, pass an isothermal curve through the point of cut-off *E*, thus obtaining the dotted line *J E L*. If there

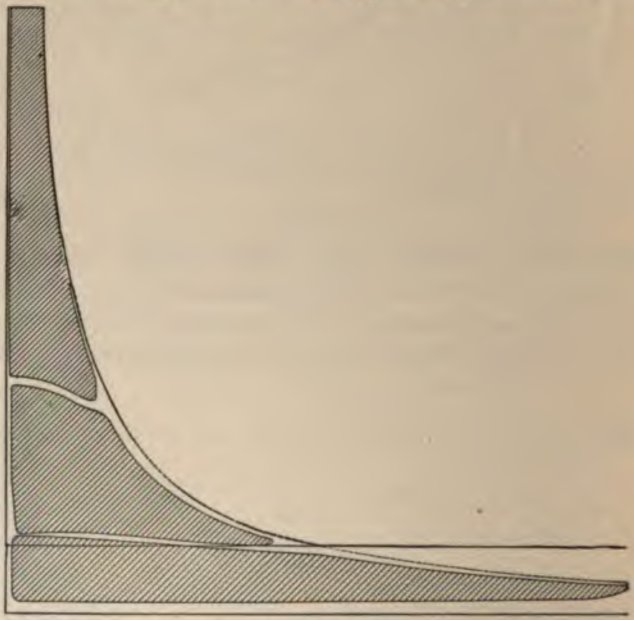


FIG. 298.

were no losses, the area $DJLNMD$ would represent the work done by the steam, and the difference between this area and the area of the shaded portions represent, approximately, the various losses. As will be seen, this loss is comparatively small, and could be made still smaller by making the cut-off in the low-pressure cylinder a trifle later.

In a manner entirely similar to that just described, the cards of a triple or quadruple expansion engine are combined. Fig. 298 shows the combined cards taken from a Reynolds-Corliss triple-expansion condensing pumping engine. The diameters of the cylinders were 28, 48, and 74 in., and the stroke was 60 in.; the clearance was 1.4%, 1.5%, and .77%, respectively.

1306. Ratio of Cylinders.—The ratio between the volumes of the two cylinders is so chosen that: 1, the power is divided as equally as possible between them; 2, the initial strains in the two cylinders may be the same; 3, the drop of pressure between the high-pressure cylinder and the receiver may be as small as possible. Numerous rules and formulas are used for this purpose. One rule is to make the number of expansions in the high-pressure cylinder 2.72. Substituting this in formula **102**,

$$E = 2.72\frac{V}{v}, \text{ or } \frac{V}{v} = \frac{E}{2.72}. \qquad \textbf{(103.)}$$

EXAMPLE.—What should be the ratio between the volumes of the two cylinders if the total number of expansions be 10?

SOLUTION.— $\frac{V}{v} = \frac{E}{2.72} = \frac{10}{2.72} = 3.68$. Ans.

Another rule is to make the required ratio equal to the square root of total ratio of expansion; that is,

$$\frac{V}{v} = \sqrt{E}. \qquad \textbf{(104.)}$$

Using the formula in the last example,

$$\frac{V}{v} = \sqrt{E} = \sqrt{10} = 3.16. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A compound engine is to be designed. The high-pressure cylinder is to be 17" × 20", with an apparent cut-off of .4, and a clearance of 10%. The total number of expansions is to be 7. What must be the size of the low-pressure cylinder having the same stroke? Ans. 30.324" × 20", say 30¼" × 20".

2. Calculate the above by formula **103.** Ans. 27½" × 20".

3. The low-pressure cylinder of a compound engine is 44" × 36". What must be the diameter of the high-pressure cylinder in order that there may be 8 expansions, with .38 cut-off and 12% clearance in the high-pressure cylinder? Ans. 23.28", say 23¼".

4. An 11" and 22½" × 18" compound engine has 13% clearance in the high-pressure cylinder. Find the point of apparent cut-off so that there may be 9 expansions. Ans. 39.5% of the stroke.

HORSEPOWER OF COMPOUND ENGINES.

1307. The actual I. H. P. of a compound or triple-expansion engine may be obtained from the indicator cards. The method used is best shown by an example.

A triple-expansion engine has the volume of its cylinders in the ratio 1 : 2½ : 6¼; that is, the low-pressure cylinder is 6¼ times as large as the high-pressure cylinder, and $\frac{6\frac{1}{4}}{2\frac{1}{2}} = 2\frac{1}{2}$ times as large as the intermediate cylinder. The low-pressure cylinder is 40" × 40". The engine makes 120 revolutions per minute. On measuring the actual cards it is found that the M. E. P. of the high-pressure cylinder is 80.5 pounds; the M. E. P. of the intermediate cylinder, 37.5 pounds, and of the low-pressure cylinder, 16.12 pounds. What is the I. H. P. of the engine?

It would be possible to calculate the I. H. P. by finding the work exerted by each cylinder separately, as if it were the cylinder of a simple engine, and then adding the works together. It is, however, easier to reduce all the pressures to the area of the low-pressure cylinder. This is done by dividing the M. E. P. of each cylinder by the ratio between its volume and the volume of the low-pressure cylinder. In the present case the M. E. P. of the high-pressure cylinder is 80.5. The volume of the low-pressure cylinder is 6.25

times that of the high-pressure cylinder, or, what is the same thing, the area of the low-pressure piston is $6\frac{1}{4}$ times that of the high-pressure piston. Therefore, to produce the same work, the M. E. P., when acting in the low-pressure cylinder, must be $\frac{1}{6.25}$ of what it was in the high-pressure cylinder. The M. E. P. of the small cylinder reduced to the area of the low-pressure cylinder is, therefore, $\frac{80.5}{6.25} = 12.88$ pounds. Likewise the M. E. P. of the intermediate, reduced to the low-pressure cylinder, is $\frac{37.5}{2.5} = 15$ pounds. The M. E. P. of the low pressure-cylinder, of course, remains the same. The total M. E. P., reduced to the low pressure-cylinder, is, therefore, $12.88 + 15 + 16.12 = 44$ pounds. Now, substituting this M. E. P., the area of the low-pressure cylinder, the length of stroke and revolutions per minute in formula **98,**

$$\text{I. H. P.} = \frac{PLAN}{33{,}000} = \frac{44 \times 40 \times 40^2 \times .7854 \times 120 \times 2}{12 \times 33{,}000} = 1{,}340.42.$$

1308. Fig. 299 shows an elevation of a **tandem compound non-condensing engine,** in which *a* is the high-

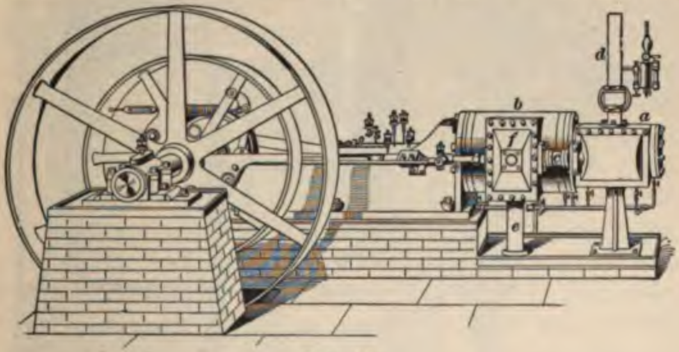


FIG. 299.

pressure cylinder, which is placed behind the low-pressure cylinder *b*. Many makers prefer to place the large cylinder behind, since it is then easier to remove the pistons and

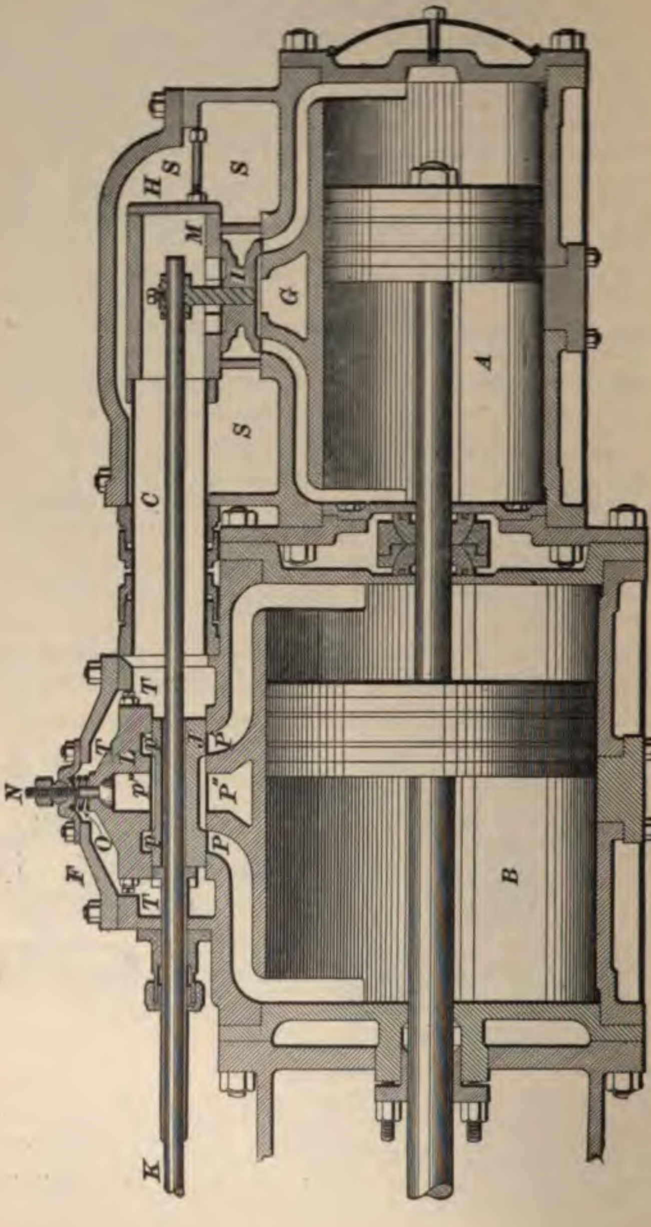


FIG. 300

examine the cylinders in case of repairs. When the small cylinder is behind, it must be entirely removed from the engine before the pistons can be taken out, while, if the large cylinder is behind, the small piston can be pulled out. The steam pipe is shown at *d*, and the exhaust pipe at *e*. After the steam has expanded in *a*, it is discharged through the connecting pipe *c* into the steam chest *f* of the low-pressure cylinder. After doing its work in *b*, it is exhausted into the condenser or atmosphere through the pipe *e*. As will be seen, a shaft governor is used. Both pistons are, of course, attached to the same piston rod.

Fig. 300 shows a horizontal section taken through the center line of the cylinders of this engine. The live steam fills the space *S*, *S*, *S*, but has no communication with the pipe *C*, which leads to the steam chest *F* of the low-pressure cylinder, being prevented from entering *C* by the plate *H*. The exhaust port *G*, however, connects directly with *C* so that the steam, after expanding in *A*, exhausts through *G* into *C* and fills the space *T*, *T*, *T*, from whence it passes to the cylinder *B*. Both valves *I* and *J* are moved by the same valve stem *K*. Since the valves present many peculiarities over those before mentioned, a detailed description of them will now be given. The ordinary **D** slide-valve is subjected to a great pressure by the steam on its back. Thus, if, in a large engine, the valve is say 10″ long by 14″ wide and the boiler pressure is 100 lb., the pressure on the back of the valve when in its central position is $10 \times 14 \times 100 = 14{,}000$ pounds, or 7 tons. This great weight causes the valve seats to wear very fast, and, as they do not wear evenly, they soon leak badly. To obviate this difficulty, what are termed **balanced valves** are employed. Of this type are the valves shown in Fig. 300. Here flat plates *L* and *M* are placed on top of the valves. The plates and valves are ground together so that they fit perfectly, and no steam can get on the back of the valves. For convenience only the valve *J* will be described in connection with the balancing. The flat plate *L* is called a **pressure plate,** and is prevented from pressing the valve with great force to its seat owing to the steam pressure on the back of the plate by reason of the bolt N. There must be some pressure upon the valve in order to keep it to its seat, and this is furnished by the spring O. The valve would still be unbalanced from beneath if the bottom of the pressure plate were a perfectly smooth surface; for, when in the position shown, the steam filling the ports P and P' presses upwards against the valve and forces it against the pressure plate. To counteract this, recesses p and p', having the same width and length as the steam ports P and P', are cut in the pressure plate and steam is allowed to enter each from the corresponding ports P and P', which are exactly opposite the recesses. For the same reason, the recess p'' balances the exhaust steam entering the port P''.

1309. By using balanced valves, another advantage is obtained besides the decreased wear; viz., the governor parts, eccentrics, etc., can be made much lighter. A pressure plate should not be used unless some provision has been made for allowing the valve to be raised from its seat in case water should be formed in the cylinder by the

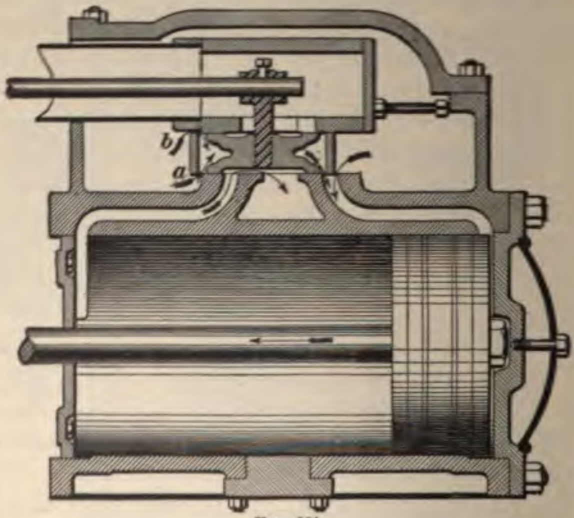


FIG. 301.

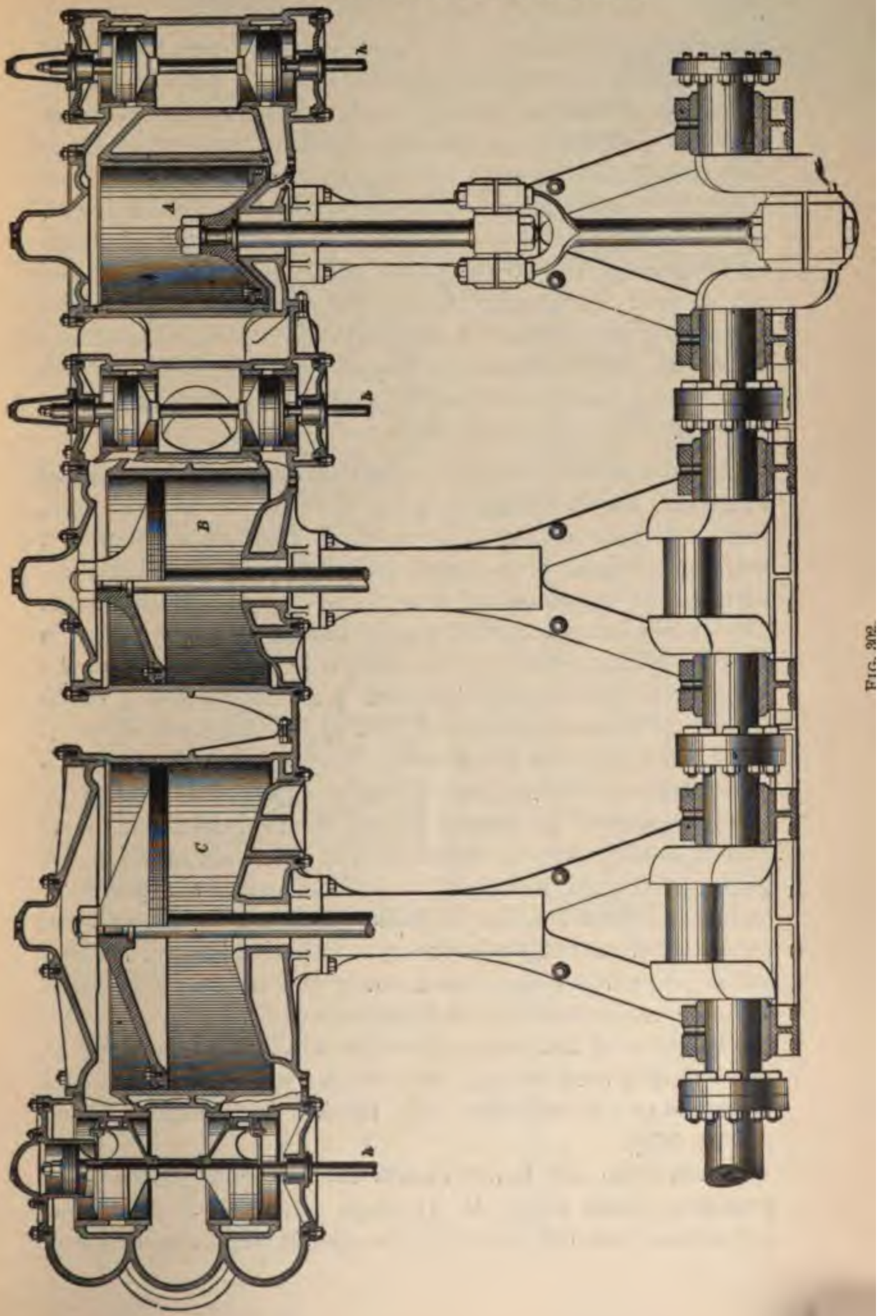


FIG. 302.

condensation of steam. In the present case, this is provided by the spring *O*, which will allow the pressure plate and valve to be raised together should water collect in the cylinder. Another peculiarity of the high-pressure cylinder valve is that it has three points or openings for the admission of steam to the cylinder. This is clearly shown in Fig. 301, which shows the valve and piston in position at the commencement of the stroke. The three points of admission are secured by making the valve hollow and allowing the steam to enter at *a* and *b*, flow through the valve and into the steam port, as shown by the arrows. The object of this is to allow a wider port opening without a corresponding increase in the travel of the valve.

Fig. 302 shows a section of a **triple-expansion marine engine.** Marine engines are always of the vertical type, since a vertical engine occupies much less floor space than a horizontal engine of the same capacity. *A* is the high-pressure, *B* the intermediate, and *C* the low-pressure cylinder. The cranks are placed 120° apart; hence, when the piston in *A* is beginning its stroke, the pistons in *B* and *C* are on the same level, one having traveled $\frac{3}{4}$ and the other $\frac{1}{4}$ of its stroke. The steam exhausts from *A* into *B*; from *B* into *C* and from *C* into the condenser. The valves (not shown in their proper positions) consist of pistons connected by a rod. They are moved by means of the stems *h* which are connected to eccentrics on the shaft *S* in the usual manner. A glance at the cut will show that these valves are perfectly balanced. The shaft *S* is made considerably larger than necessary for strength in order to obtain a greater bearing surface and thus reduce the amount of wear due to friction. It is also made hollow to reduce the weight.

The route of the steam through a triple engine may be more clearly seen in Fig. 303, which represents a plan and elevation of the cylinders of a marine engine (not the one of Fig. 302).

Steam from the boiler enters the two ends of the high-pressure steam chest *H*, through the pipe *A A'*. It is exhausted from the center of the steam chest through *B* to

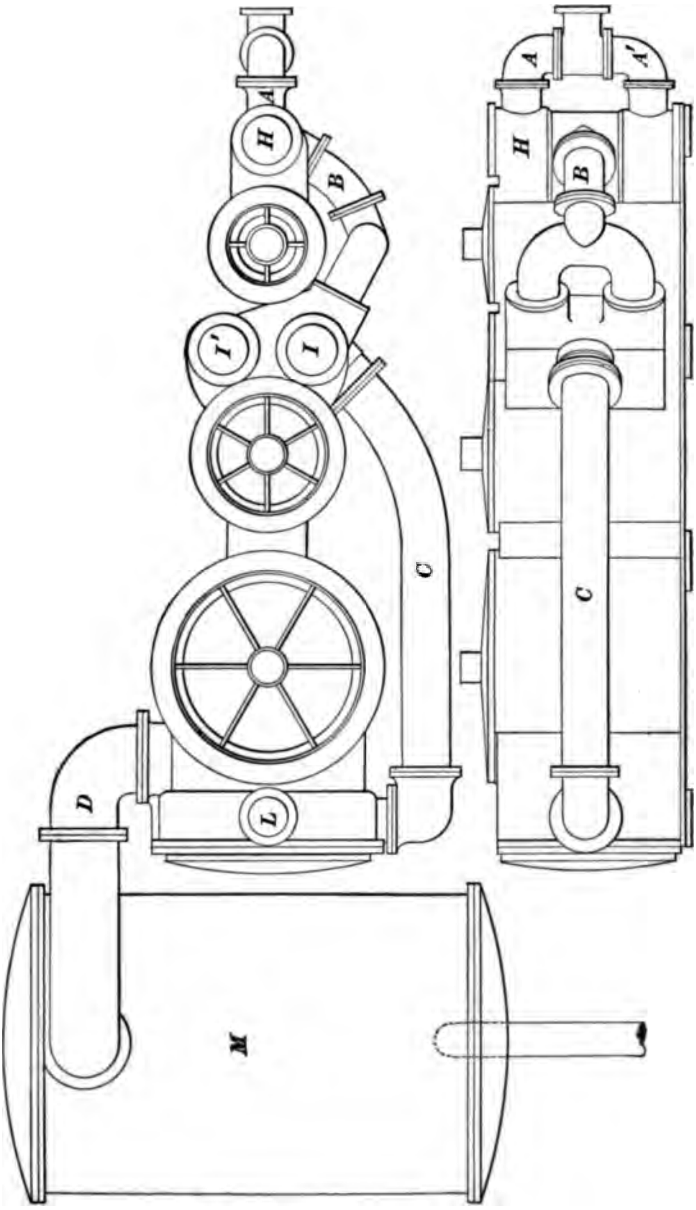


FIG. 308.

the two ends of the intermediate steam chest $I\ I'$. After expanding in the intermediate cylinder, it exhausts from the middle of I through the pipe C into the low-pressure steam chest L; and finally from L through the pipe D into the condenser M. After condensing in M, the steam, in the form of water, is pumped out of the condenser into the hot well, and from there into the boiler.

It may be observed that the steam chest I has two piston valves. In large triple-expansion engines, each cylinder may have two or even more piston valves.

MECHANICS OF THE STEAM ENGINE.

1310. **In Fig. 304, let $O\ C$ represent the crank; then, the circle $C'\ C''\ M$ represents the path of the crank-pin.**

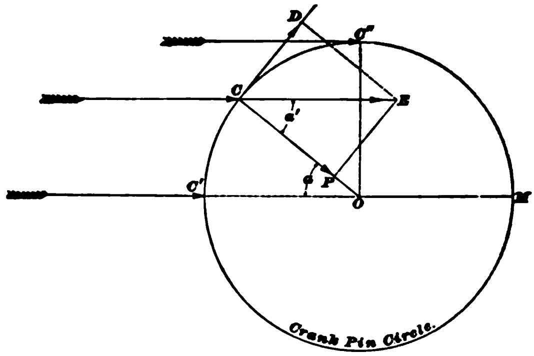


FIG. 304.

The steam exerts, through the piston, piston rod, and connecting-rod, a pressure on the crank-pin C. Let it be assumed that the connecting-rod is very long, so that the direction of the pressure on the pin is always horizontal.

Now, when the crank is in the position $O\ C'$, the horizontal pressure of the steam simply produces a pressure on the bearing of the crank-shaft; there is no tendency whatever to turn the crank around O as a center, and it is, therefore, said to be on the dead center. When the crank is in the position $O\ C''$ *all* the pressure of the steam on the piston is expended in turning the crank around O as a center, and there is no pressure on the bearing. This is because the direction of the pressure at this point is at right angles to the crank, or, in other words, tangent to the crank circle.

When the crank is in some other position, as $O\ C$, there will be a tendency to turn the crank around O as a center, and also to produce a pressure on the crank-shaft bearing. To find the magnitudes of these forces tending to rotate the crank and tending to produce a pressure on the bearing, proceed as follows: Let $C\ E$ represent to some scale the horizontal pressure of force on the crank-pin at C. The turning force acts in the direction of the tangent $C\ D$, while the force which produces the pressure on the bearing acts along the crank, or in the direction $C\ O$. The force $C\ E$ may be resolved in the two directions $C\ D$ and $C\ O$ by means of the parallelogram of forces. From E, draw $E\ P$ parallel to $C\ D$ and $E\ D$ parallel to $C\ O$. Then, $C\ D$ is the tangential or turning force, and $C\ P$ the force producing pressure on the bearing, both to the same scale as $C\ E$.

Let a be the angle $C\ O\ C'$ which the crank makes with the horizontal. $C\ E$ and $C'\ O$ are parallel, and $C\ P$ and $C\ O$ are parallel (coincide); hence, by geometry, $C\ O\ C'$ and $P\ C\ E$ are equal, or angle a = angle a'.

Tangential force $= C\ D = E\ P = C\ E \sin E\ C\ O = C\ E \sin a$, or *the force tending to turn the crank is equal to the horizontal force on the crank-pin multiplied by the sine of the angle which the crank makes with the horizontal.*

When the crank is at $C'\ O$, a is 0; therefore, $\sin a$ is 0 and the tangential force is 0, as it should be. When the crank is at $O\ C''$, a is 90°, $\sin a$ is 1, and the tangential force is the same as the horizontal force.

The radial force, or the force which produces pressure on the bearing, may be shown in the same manner to be equal to the horizontal force multiplied by the cosine of the angle which the crank makes with the horizontal, thus,

Radial force $= C\ P = C\ E \cos E\ C\ O = C\ E \cos a$.

EXAMPLE.—The horizontal force on the crank-pin is 6,000 pounds. What will be the tangential and radial forces when the crank makes an angle of 60° with horizontal direction of the force?

SOLUTION.—$a = 60°$; $\sin a = .866$; $\cos a = .5$.

Tangential force $= 6{,}000 \times \sin a = 6{,}000 \times .866 = 5{,}196$ lb.

Radial " $= 6{,}000 \times \cos a = 6{,}000 \times .5 = 3{,}000$ lb.

1311. A diagram showing the tangential pressure for every position of the crank may be easily constructed, as

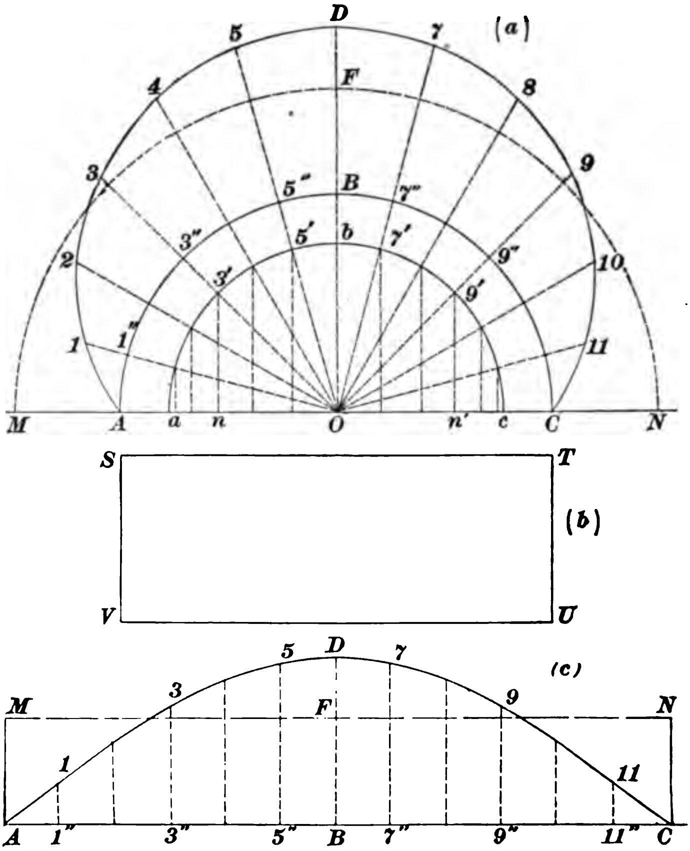


FIG. 305.

follows: For simplicity, assume the piston pressure to be constant throughout the stroke, as shown by the indicator diagram or card, Fig. 305 (*b*); suppose, also, that the connecting rod is very long, so that the direction of pressure is always horizontal. Let the length of the crank be $O A$; then, with O as a center and $O A$ as a radius, describe one-half the crank-pin circle $A B C$. Let $O a$ represent the uniform pressure $s v$ and describe the semicircle $a b c$. Divide $a b c$ or $A B C$ into a convenient number of equal parts (12 is most convenient), and through each division draw the radial lines $O 1$, $O 2$, $O 3$, etc., prolonging them beyond $A B C$. From the points where these radial lines intersect the semicircle $a b c$ drop perpendiculars on line $a c$. Then, each perpendicular represents the tangential component of the pressure when the crank is in that position. For example, when the crank is at $O 3'$, the length $3' n$ represents the tangential pressure, since

$$3'n = O\,3' \sin 3'O\,n = O\,a \sin 3'O\,n = \text{horizontal pressure} \times \text{sine of crank angle.}$$

Now, lay off these perpendiculars, each on its own radial line outwards from the crank-pin circle $A B C$; that is, lay off $3'n$ on the radial line $O 3$, the length $3\text{-}3''$ being made equal to the length $3'n$.

We thus obtain the series of points *1*, *2*, *3*, etc.; the curve $A D C$, drawn through these points, will represent the tangential pressures for all points of the stroke.

It will be noticed that at A and C, the dead points, the tangential pressure is 0, and at D it is equal to the horizontal pressure.

In Fig. 305 (*c*), the tangential pressure diagram is represented with a straight base. The semicircle $A B C$ has been straightened out, the ordinates $1\text{-}1''$, $2\text{-}2''$, etc., remaining the same as before.

Since the ordinates of (*c*) represent the tangential pressures on the crank to the same scale that the ordinates of (*b*) represent pressures on the piston, and since the length $A C$ represents the distance passed through by the crank to the same scale that $V U$ represents the distance passed through by the piston, it follows that the area of (*c*) represents the work done by the crank-pin during a half revolution of the crank, or during one stroke of the piston.

The work done on the piston by the steam must be equal to the work given up by the crank-pin. Therefore, since (*b*) and (*c*) have the same scale of pressures and distances, since (*b*) represents the work done on the piston and (*c*) represents the work done by the crank-pin, their areas must be equal; and so they will be found to be by actual measurement.

The "mean ordinate" of (*c*) may now be found from the above considerations. The length of the semicircle $A\ B\ C$ of (*a*) is $\frac{\pi}{2}$ times the length of the diameter $A\ C$; but the base $A\ B\ C$ of the diagram (*c*) is equal in length to the semicircle $A\ B\ C$, and the length $V\ U$ of (*b*) is equal to the diameter $A\ C$, since both represent the length of stroke. Therefore, the base $A\ B\ C$ of (*c*) is $\frac{\pi}{2}$ times as long as the base $U\ V$ of (*b*). The area of (*b*) and (*c*) are the same; consequently, the mean ordinate of (*b*) must be $\frac{\pi}{2}$ times that of (*c*), or $S\ V = \frac{\pi}{2} A\ M$; therefore, $A\ M = \frac{2\ S\ V}{\pi}$.

The mean ordinate of the diagram of tangential pressures on the crank-pin is always $\frac{2}{\pi}$ *times the mean ordinate or M. E. P. of the indicator card.*

In both (*a*) and (*c*), $M\ F\ N$ is the line of average tangential pressures, and in both cases is drawn parallel to $A\ B\ C$ at a distance from it equal to the mean ordinate $A\ M$.

The case just described is much simpler than is ever met with in practice. The pressure of the steam is rarely constant throughout the whole stroke, and the connecting rod is never so long that the pressure exerted by it on the crank pin may be regarded as always horizontal.

1312. In order to apply the foregoing principles to an actual case, it is necessary to use the net pressure (forward pressure, less the back pressure on the other side of the piston) on the piston during one-half a revolution. This is easily done by combining the two curves *A B C D* and *K L C M*, Fig. 267 (which represent the steam pressure on opposite sides of the piston during the forward stroke), as shown in Fig. 306, where *A B a c* represents *A B C D*, and *e f a b* represents *K L C M*. Joining *A* and *e* and *b* and *c* by straight lines, the figure is completed. Any ordinate, as *g h*, measured to the scale of the indicator spring, is the net pressure on the piston urging it ahead when it occupies the position of *h* of its stroke. The net pressure is 0 at *a*, i. e., the pressure is the same on both sides of the piston. Between *a* and *b c* the net pressure is negative, or, in other words, the back pressure is greater than the forward pressure, and the piston is carried to the end of its stroke solely by the energy stored in the fly-wheel.

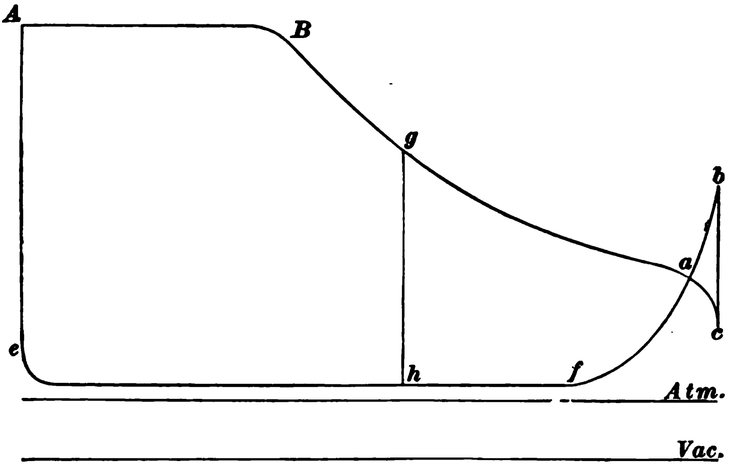


FIG. 306.

1313. Let us now take a diagram of *net* pressures on the piston like the one in the figure just shown, and, assuming the connecting-rod to be four times the length of the crank, work out the corresponding diagram of tangential pressures.

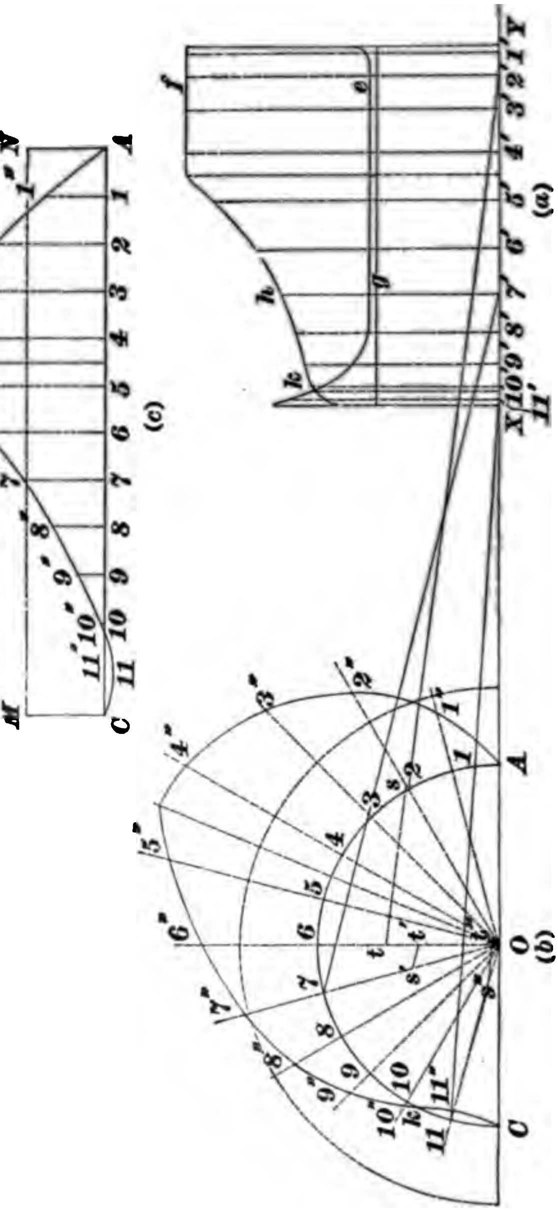

(a)
X
Y
1' 2' 3' 4' 5' 6' 7' 8' 9' 10' 11'
f
e
g
h
k
(b)
O
A
C
1 2 3 4 5 6 7 8 9 10 11
1" 2" 3" 4" 5" 6" 7" 8" 9" 10" 11"
s
t
s'
t'
k
(c)
M
N
C
A

FIG. 307.

The diagram or card is shown at (*a*) Fig. 307. At any convenient distance below the card, draw a line *C Y* parallel to the atmospheric line of the cards. Project the two ends of the card down on *C Y*, locating the points *X* and *Y*. Lay off, on the line *C Y*, $Y A = X C = 2 X Y = 4 O A =$ the length of the connecting-rod. Take the point *O* midway between *A* and *C*, and with *O A* as a radius, strike the semicircle *A 6 C*. *X Y* now represents the length of the stroke to the scale of the diagram, and *A 6 C* the path of the crank-pin. Divide *A 6 C* into a convenient number of equal parts, and through the points of division draw the radial lines *O 1*, *O 2*, *O 3*, etc., prolonging them some distance beyond the semicircle. Now find the positions of the piston corresponding to the crank positions *O 1*, *O 2*, etc. This may be done by taking the length *A Y* of the connecting-rod with a pair of dividers; then, placing one leg of the dividers on *1*, *2*, *3*, etc., in succession, strike off the piston positions *1′*, *2′*, *3′* on the line *X Y*. The tangential pressure may be found in the following manner: Suppose it is desired to find the tangential pressure on the crank when it is in the position *O 2*. The pressure on the piston at this point is shown on the card to be *e f*. Lay off from *O* the length *e f* on the radial line *O 2*, thereby obtaining the length *O s*. Draw in the connecting-rod *22′*, and through *s*, draw *s t* parallel to *22′* and intersecting the vertical line *O 6″ t*. Then, *O t* is the tangential pressure to the same scale that $e f = O s$ is the pressure on the piston. When the crank is at *7* the piston is at *7′*, and the pressure on it from the card is *g h*. *O s′* is laid off on *O 7* equal to *g h*, *s′ t′* is drawn parallel to *7 7′*; then, *O t′* is the tangential pressure on the crank when it is in the position *O 7*. In this manner, the tangential pressure can be found for all positions of the crank, and, being laid off radially from the crank-pin circle, as in the previous case, they form the curve *A 4″ 7″ k C*. At *k*, the pressures on both sides of the piston are equal, and, consequently, the tangential pressure is zero. The point *k* in (*b*) is determined by dropping a perpendicular from *k* to *X Y*, and with the point of intersection as a center and a radius equal to the length of the connecting-rod, describe an arc cutting *C* 6 *A* in *k*. Beyond *k*, the back pressure exceeds the forward pressure, and the tangential pressure must be laid off below the semicircle *C k* 6 *A*, as shown. As before, the semicircle *A* 6 *C* may be laid out straight, and the diagram shown in (*c*) obtained. The area of the latter will again be found to be equal to the area of the diagram (*a*). The line *M N*, as before, represents the average tangential pressures on the crank-pin. It is usually a good plan to find the crank and piston positions at cut-off and get the tangential pressure at that point, since a sharp change in the curve will there occur, as shown in the figure.

1314. The load on the engine, or the resistance against which the engine works, is nearly always constant. That is, it requires a practically constant force to drive the shafting or machines. It is, therefore, very desirable that the force turning the crank-shaft should be as nearly constant as possible. It has been shown by the diagrams of Figs. 305 and 307 that this tangential force is very far from being constant; that at the dead points it is zero and near the middle of the stroke it is greatest. It can now be shown why compound and duplex engines, which have their cranks at right angles, possess advantages over the simple engine. In Fig. 308 is shown a diagram for a cross-compound engine with cranks at right angles. When one crank is beginning its stroke, the other is at the middle of the stroke, and vice versa. Therefore, when the curve of one crank touches the base line, as at *m*, the other will be at or near its highest position, as at *n*. Now, adding the corresponding ordinates of the two curves together, we obtain the dotted curve

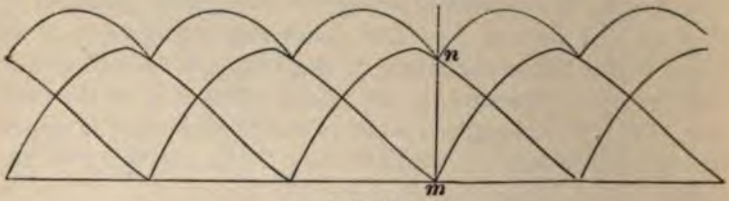


FIG. 308.

which, therefore, represents the total tangential pressure tending to turn the crank-shaft. It is apparent at a glance how much more nearly constant is the tangential pressure of the compound engine diagram as compared with the tangential pressure of the simple engine diagram. Consequently, with the same steam pressure and weight of fly-wheel, the compound or duplex engine with crank at 90° will run more steadily than the simple engine, while a triple-expansion engine with 3 cranks, making angles of 120° with each other, will run steadier than the compound.

1315. A tandem compound engine also has a mechanical advantage over the simple engine, as a little consideration will show. It was proven in Art. **1301** that a compound engine was equivalent to a simple engine whose cylinder was of the same dimensions as the low-pressure cylinder, and which expanded its steam the same number of times as the compound. For convenience of illustration, assume the cut-off in the high-pressure cylinder to be $\frac{1}{2}$, and that the ratio of the volumes of the two cylinders is 1 : 4; that there is no clearance and no receiver; that there is no loss of pressure due to wire-drawing, friction, etc., and, finally, that the steam is carried full stroke in the low-pressure cylinder. The total number of expansions is 8, 2 in the small cylinder and 4 in the large cylinder. Assume the steam pressure to be 120 pounds, absolute, and the absolute back pressure to be 15 pounds. Denote the area of the large piston by A; then, the area of the small piston is $\frac{A}{4}$. The terminal pressure in the small cylinder is, evidently, 60 pounds, and this is the initial forward pressure in the large cylinder and the initial back pressure in the small cylinder.

First consider the simple engine. The initial forward pressure is 120 pounds and the back pressure 15 pounds; hence, the net pressure urging the piston ahead is $120 - 15 = 105$ pounds. The total force acting on the piston is $105\,A$.

Considering the compound engine, the initial forward pressure in the small cylinder is 120 pounds; the back pressure 60 pounds, and the net pressure $120 - 60 = 60$ pounds. Since the area of the small piston is $\frac{A}{4}$, the total force tending to drive the small piston forward is $60 \times \frac{A}{4} = 15\ A$. The initial forward pressure in the low-pressure cylinder is 60 pounds, the back pressure is 15 pounds, and the net pressure urging the piston ahead is $60 - 15 = 45$ pounds. The total force tending to drive the large piston forward is $45\ A$. The total initial force acting on both pistons of the compound is $45\ A + 15 A = 60\ A$; in the simple engine it was $105\ A$. Since the greatest strains occur when the forces which produce them are greatest, it is evident, from the above, that the various parts of a compound engine (connecting-rod, crank, shaft, etc.) will not need to be made so large as in a simple engine having the same mean effective pressure, and the volume of whose cylinder equals the volume of the low-pressure cylinder. This is, however, a disadvantage when the engine has a very heavy load to start, as in the case of a locomotive, for in that case the compound engine might not be able to start, although it could keep itself in motion after it had once been started.

CONDENSERS.

1316. It has been shown that the thermal efficiency of the steam engine, $\frac{T_1 - T_2}{T_1}$, may be increased by either raising the temperature, T_1, of the live steam, or by lowering the temperature, T_2, of the exhaust steam. T_1 may be raised by increasing the boiler pressure of the steam; T_2 may be lowered by using a **condenser.**

In non-condensing engines—that is, engines which are not supplied with a condenser—the steam is exhausted into the atmosphere, and therefore the exhaust steam must have, at least, the pressure of the atmosphere; in practice the back pressure of steam in a non-condensing engine is scarcely ever less than 16 pounds above vacuum, and is oftener 17 pounds or more. In good condensing engines the back pressure is often as low as 2 pounds above vacuum.

1317. Suppose that the boiler pressure of the steam is 80 pounds absolute, the temperature corresponding to the pressure is, from the steam table, 311.9° F., and the absolute temperature is, therefore, $460° + 311.9° = 771.9°$ F. The absolute temperature corresponding to a pressure of 17 pounds is $460° + 219.5° = 679.5°$ F., and corresponding to a pressure of 3 pounds is $460° + 141.7° = 601.7°$ F. The thermal efficiency of the engine, if non-condensing, is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 679.5}{771.9} = 12\%$, nearly; if condensing to 3 pounds absolute, the efficiency is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 601.7}{771.9} = 22\%$.

1318. The increase of economy by the use of the condenser may be shown in another manner. Let $A\,B\,C\,D\,E\,F$, Fig. 309, be an indicator card from a non-condensing engine. $M\,N$ is the atmospheric line and $O\,X$ the vacuum line. The

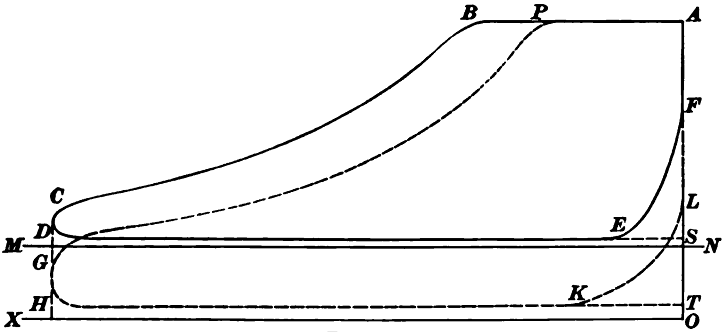


FIG. 309.

back pressure, as shown by the card, is $O\,S$. The area of the card represents to some scale the work done per stroke. Now let a condenser be attached to the engine. The back pressure will be lowered to $O\,T$, the line $H\,K$, instead of

D E, now being the lower line of the card, and *A B C H K L* will be the new card, its area, as before, representing the work done per stroke. Hence, by adding a condenser to the engine, the work per stroke has been increased by an amount represented by the area *F E D H K L*, the steam consumption remaining the same. Suppose the steam to be cut off at a point *P*, making the area of the card, *A P G H K L*, equal to the area of the original card, *A B C D E F*. Then, the work per stroke is the same in both engines, but the condensing engine uses an amount of steam per stroke represented by the length *A P*, while the non-condensing engine uses an amount represented by *A B*. Either case shows the economy of the condenser.

1319. There are two types of condensers in general use: The **surface condenser** and the **jet condenser.**

In the former, the exhaust steam comes in contact with a large area of metallic surface, which is kept cool by contact with cold water. In the latter, the exhaust steam, on entering the condenser, comes in contact with a jet of cold water. In either case, the entering steam is condensed to water, and, in consequence, a partial vacuum is formed. If a sufficient amount of cold water was used, the steam on entering would instantly condense, and a practically perfect vacuum would be obtained, were it not for the fact that the feed water of the boiler always contains a small quantity of air, which passes with the exhaust steam into the condenser, and therefore partially destroys the vacuum. To get rid of this air, the condenser is fitted with an air pump, which pumps out both the air and the water, into which the steam condenses.

1320. The Surface Condenser.—Fig. 310 is a perspective view of the Wheeler surface condenser. Fig. 311 is a sectional view of the same. The cold condensing water is drawn from some water supply through *M*, and forced by the circulating pump *Q* into the inlet *C* of the condenser. From *C* the water is forced to the chamber *F*, and flows, as indicated by the arrows, through the inner tubes of the lower layer of double tubing to the left, and having passed through their entire length, it returns through the space between the outside of the inner and inside of the outer tubes into the chamber *G*. Fig. 312 shows more clearly the arrangement of this "double tubing." From *G* (Fig. 311) it passes through *E* to *H*, and from *H* to *I* through the upper layer of double tubing, as has already been explained. From *I*, it is discharged through the nozzle *D*, carrying with

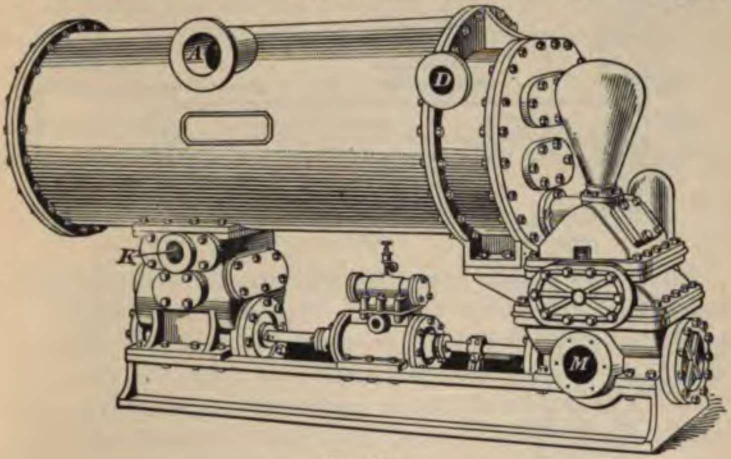


FIG. 310.

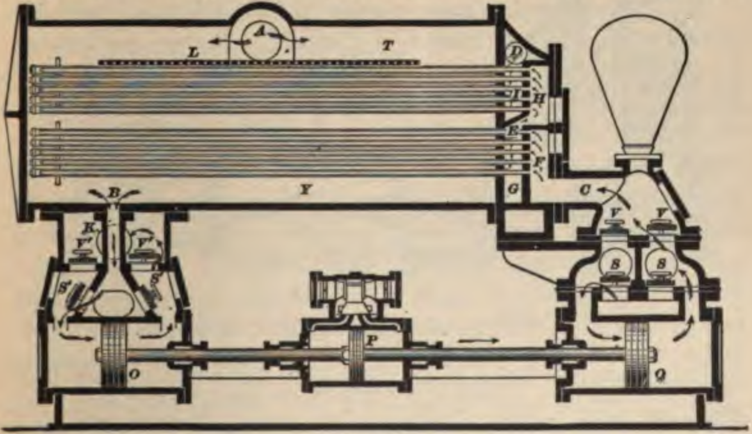


FIG. 311.

it all the heat it has received by coming in contact with the two layers of double tubing.

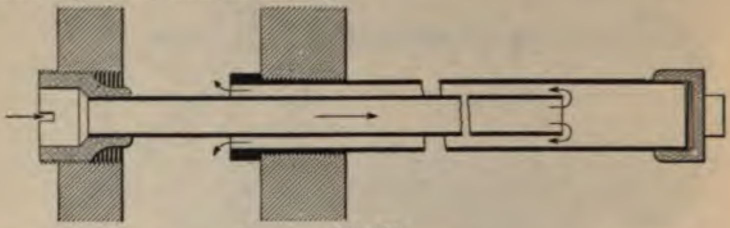


FIG. 312.

The nozzle at *A* is connected with the exhaust pipe of the steam cylinder of an engine. The movement of the air pump piston *O* draws air through the orifice *B* from the condenser cylinder and discharges it through the valves and nozzle *K* in a manner clearly indicated by the arrows. The valves *S'* and *V'* are opened and closed automatically by the pressure of the air beneath them and by the pressure of the air and springs above them. A partial vacuum is created in the condenser cylinder *Y* by the action of the air pump, thus sucking the condensed steam from the engine cylinder into the condenser cylinder.

As the exhaust steam enters the condenser cylinder through *A*, it first comes in contact with the perforated scattering plate *L*, which protects the upper tubing from the damaging effect of direct contact with the exhaust steam. The steam comes in contact with the cold tubes, through which the cold water is being pumped, and condenses. As soon as this occurs, the condensed exhaust steam collects at the bottom of the condenser cylinder and runs through *B* into the air pump cylinder, from which it is discharged while still heated and made use of as boiler feed-water. The temperature of this condensed exhaust steam boiler feed-water will be less as the vacuum in the condenser cylinder is more complete. The advantage of making use of this condensed exhaust steam as boiler feed-water will be clearly seen when it is remembered that it has a higher temperature than the ordinary feed-water, and will, therefore, require less fuel in converting it into steam.

In this condenser, the circulating and air pumps are run by an independent steam cylinder *P*. They are often connected directly to the main engine, and have their motion imparted to them by some of its moving parts, generally the crank-shaft.

1321. The Jet Condenser.—In Fig. 313 is shown a section of a Worthington independent jet condenser. The cold water enters the condenser at *B*, passes down the spray pipe *C*, and is broken into a fine spray by the cone *D*. The exhaust steam in the meantime comes in at *A*, and, mingling with the spray of cold water, is rapidly condensed. The velocity of the entering steam is imparted to the water, and

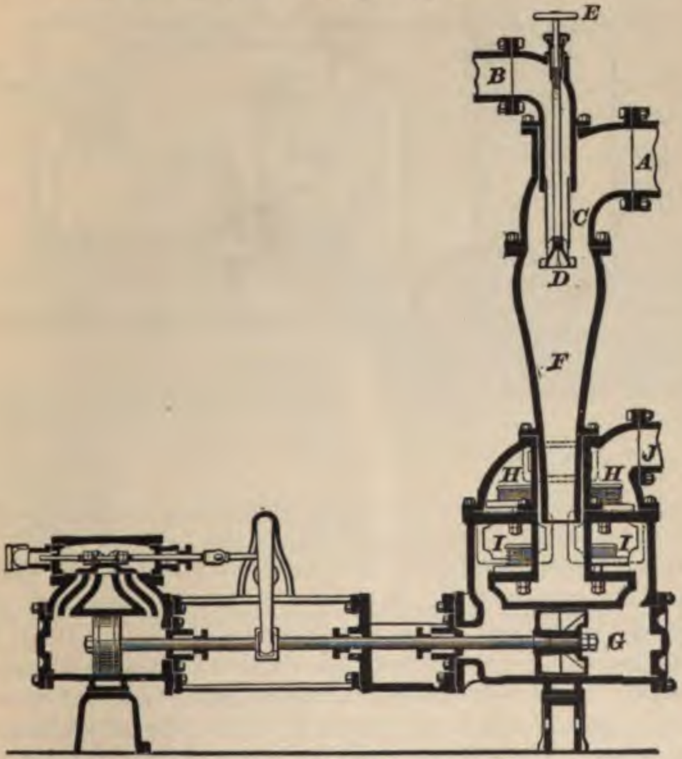


FIG. 313.

the whole mixture of steam, water, uncondensed vapor, and air is carried with a high velocity through the cone *F* into the pump cylinder *G*, whence it is forced by the pump through the discharge pipe *J*.

1322. In Fig. 314 is shown a jet condenser in connection with the boiler and engine. The exhaust pipe *A* leads directly to the condenser. The injection pipe *B* draws water from the reservoir *C*. After the steam is condensed, the mixture of exhaust steam and injection water is discharged through *D* into the sewer. A portion of this discharge, however, flows through *E* to the feed pump *G*, which forces it through the coil in the heater *F* to the pipe *H* leading to the boiler. The exhaust from the two pumps is discharged into the feed-water heater through the pipe *M*. It will be noticed that water from the overflow pipe *D* enters the feed pump under a slight head. This is because the water is heated by the exhaust steam, and hot water can not be raised by a pump like cold water. *N* is a pipe leading from the boiler, and supplies steam for both pumps.

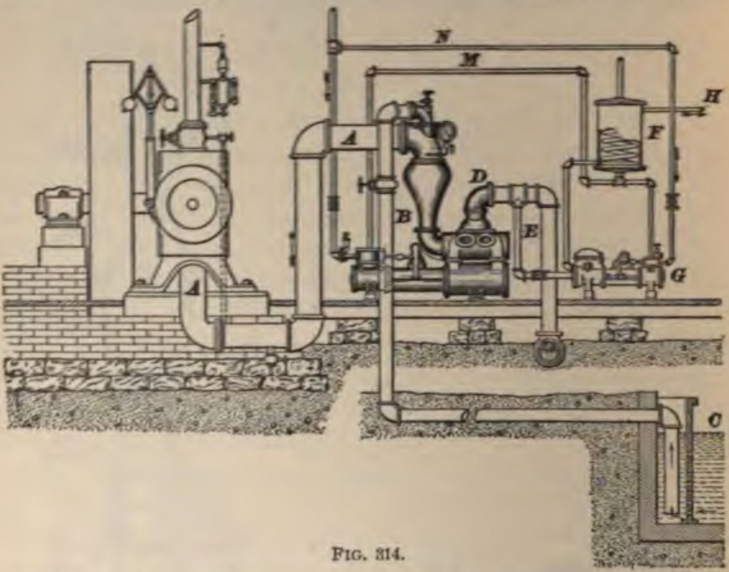


FIG. 314.

1323. The **water required by a condenser** may be calculated as follows:

Let t_1 = the temperature of departing condensing water;
t_2 = " " entering " "
t_3 = " " the condensed steam upon leaving the condenser;
H = total heat of one pound of steam at the pressure of the exhaust;
W = the weight of water required per pound of steam condensed.

The heat given up by the one pound of steam condensing to the temperature t_3 is $H - (t_3 - 32°)$ B. T. U.; the heat absorbed by the water is $W(t_1 - t_2)$. Therefore,

$$W(t_1 - t_2) = (H - t_3 + 32), \text{ or } W = \frac{H - t_3 + 32}{t_1 - t_2}. \qquad \textbf{(105.)}$$

EXAMPLE.—The steam exhausts into the condenser at a pressure of 4 pounds absolute. The temperature of the condensing water on entering is 60°, and on leaving 100°. The temperature of the condensed steam on entering the air pump is 140°. How many pounds of condensing water are required per pound of steam?

SOLUTION.—From the steam table, the total heat of a pound of steam at 4 pounds pressure above vacuum is 1,128.641 B. T. U.

Then, $W = \frac{H - t_3 + 32}{t_1 - t_2} = \frac{1{,}128.641 - 140 + 32}{100 - 60} = 25.16$ pounds. Ans.

In the jet condenser the final temperatures of the cooling water and the condensed steam are the same—that is, $t_3 = t_1$.

EXAMPLE.—Steam exhausts into a jet condenser at a pressure of two pounds above vacuum. The temperature of the condensing water is 60°, and the temperature of the mixture as it enters the pump is 135°. How much water must be used per pound of steam?

$W = \frac{H - t_3 + 32}{t_1 - t_2} = \frac{1{,}120.462 - 135 + 32}{135 - 60} = 13.566$ pounds. Ans.

1324. The surface condenser is more generally used than the jet condenser, especially in marine practice. They cost more originally and require more condensing water, but they possess the great advantage of allowing only the condensed steam to return to the boiler; hence, any water, no matter how impure, may be used for condensing. The jet condenser, on the other hand, pumps both condensing water and condensed steam into the hot well, and hence, if impure water be used, it will find its way into the boiler and form a scale.

1325. Cooling Surface.—The cooling surface required by a surface condenser may be calculated as follows:

Let $S =$ the required surface in square feet;

$W =$ total weight of steam used per hour;

$$\text{Then, } S = .0944\, W. \quad \textbf{(106.)}$$

EXAMPLE.—What cooling surface should be given the surface condenser of an engine developing 240 I. H. P. and using 23 pounds of steam per I. H. P. per hour?

SOLUTION.—Using formula **106,**

$$S = .0944\, W = .0944 \times 240 \times 23 = 521.1 \text{ sq. ft.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. How many gallons of water are needed per hour to condense the steam from a 150 horsepower engine which uses 31.63 pounds of steam per horsepower per hour? The pressure of the exhaust is 4 pounds absolute; the temperature of the condensed steam is 172°; of the water, as it enters the condenser, 66°, and on leaving, 112°.

Ans. 12,204.5 gal. per hour.

2. A jet condenser is attached to a direct-acting steam pump which runs at a piston speed of 111 feet per minute. The steam cylinder is 28" × 36" and the steam pressure is 70 pounds gauge. How much water must be furnished per hour to the condenser, if its temperature on entering is 54° and on leaving 160°, the vacuum gauge showing 20"?

Ans. 6,352.81 gal. per hour.

NOTE.—The steam is never cut off in a simple, direct-acting pump.

3. What cooling surface is necessary for an engine developing 360 I. H. P. and using 28 pounds of steam per I. H. P. per hour?

Ans. 951.552 sq. ft.

FLY-WHEELS.

1326. The office of the *fly-wheel* is somewhat similar to that of the governor, since each is used to obtain regularity of speed. It is the duty of the governor to adjust the effort of the engine to any large or permanent variation of the load, such as would be caused by throwing the machinery in or out of gear. It is the duty of the fly-wheel, on the other hand, to adjust the effort of the engine to sudden fluctuations of the resistance, which may occur during a single stroke of the engine. It is also the duty of the fly-wheel to equalize the varying tangential effort on the crank pin by storing energy while the piston is in the middle of the stroke, where the crank effort is greater than the resistance, and restoring it when the crank is at the dead point, where the tangential effort is zero.

1327. In Fig. 315, let *A C B* represent the tangential pressures on the crank-pin during one stroke of the piston. *A B* is the length of the semi-circumference of the crank-pin circle; *A M* is the mean ordinate, and *A B N M* represents the constant resistance to the turning of the crank. The effort and resistance must be equal; therefore, area *A C B* = area *A B N M*. It follows, then, that area *C D E* = area *A M D* + area *B N E*. At *A* the tangential effort is zero, since the crank is at a dead point. From *A* to *S* the tangential effort remains less than the resistance. The work done on the crank during that period is represented by the area *A D S*, while the resistance is represented by the area *A M D S*. Hence, the resistance is greater than the effort by the area *A M D*. At *S* the effort and resistance are equal. From *S* to *T* the effort is greater than the resistance by an amount represented by the area *D C E*. At *T* the effort is again equal to the resistance, and for the remainder of the stroke it again becomes less than the resistance.

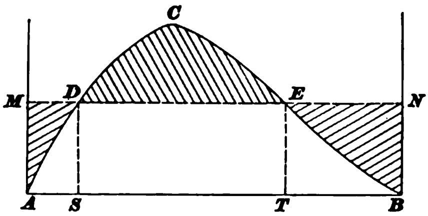


FIG. 315.

From *A* to *S* the work done by the steam was shown to be less than the resistance to be overcome; therefore, the work represented by the area *A M D* must have been obtained from some of the moving parts of the engine. The kinetic energy of a moving body is $\frac{Wv^2}{2g}$. In order to give up energy, the v of the above expression must be diminished. That is what actually takes place. All of the moving parts of the engine, the reciprocating parts, shaft, and fly-wheel, slow down a little, and, in so doing, give up enough of their kinetic energy to overcome the resistance and carry the engine past the dead center. From S to T the effort is greater than the resistance, and, consequently, the surplus energy represented by the area $C\,D\,E$ is stored up in the moving parts—that is, their velocity is increased during the part of the stroke in question, and with it their kinetic energy. For the remainder of the stroke the moving parts again slow down and give up enough energy to overcome the excess resistance $B\,N\,E$.

1328. The **weight of the fly-wheel** may be found in the following manner:

Let V_1 = the greatest velocity of the crank-pin in feet per second;

V_2 = the least velocity of the crank-pin in feet per second;

V_0 = average velocity of crank-pin in feet per second;

W = required weight of fly-wheel in pounds;

H = the number of foot-pounds per square inch of piston represented by the area $C\,D\,E$;

A = area of piston in square inches;

n = ratio between radius of fly-wheel and length of crank;

$E = \frac{V_1 - V_2}{V_0}$ = coefficient of unsteadiness.

The average velocity V_0 is known, since it is $\frac{\pi}{2}$ times the piston speed in feet per second. V_1 and V_2 are assumed so that the fraction $\frac{V_1 - V_2}{V_0} = E$ shall not exceed a certain value.

The following values of E agree well with ordinary practice:

Pumping engines, $\frac{1}{20}$.
Engines driving machine tools, $\frac{1}{35}$.
" " textile machinery, $\frac{1}{40}$.
" " spinning machinery, $\frac{1}{50}$ to $\frac{1}{100}$.
" " electric machinery, $\frac{1}{125}$ to $\frac{1}{150}$

H can be found directly from the diagram by multiplying the area $C\ D\ E$ in square inches by the vertical scale of pressures and the horizontal scale of distances. A is, of course, known, and n may be assumed at pleasure.

At the point D the crank-pin has the velocity V_2; since the radius of the fly-wheel rim is n times the length of crank, the velocity of the rim at D must be $n\ V_2$. Likewise, at E the crank-pin will have its greatest velocity V_1, and the fly-wheel rim the velocity $n\ V_1$.

The kinetic energy of the fly-wheel rim at D is $\frac{W(n V_2)^2}{2g}$; and at E it is $\frac{W(n V_1)^2}{2g}$. The kinetic energy stored up in passing from D to E is, therefore, $\frac{W n^2 V_1^2}{2g} - \frac{W n^2 V_2^2}{2g} = \frac{W n^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right)$ foot-pounds. But this kinetic energy stored up is equal to the work represented by the area CDE. Hence,

$$\frac{W n^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = A \times H. \quad (a).$$

But $\qquad \frac{V_1 + V_2}{2} = V_0$, the average velocity,

and $\qquad \frac{V_1 - V_2}{V_0} = E;$

therefore, $\left(\frac{V_1 - V_2}{V_0}\right)\left(\frac{V_1 + V_2}{2}\right) = \frac{V_1^2 - V_2^2}{2 V_0} = E V_0.$

or $\qquad \frac{V_1^2 - V_2^2}{2} = E V_0^2.$

Substituting this in (a),

$$\frac{W n^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = \frac{W n^2 E V_0^2}{g} = A H.$$

Whence, $W = \frac{A H g}{n^2 E V_0^2}$. $\qquad$ **(107.)**

EXAMPLE.—Given a single cylinder engine making 100 revolutions per minute, with a stroke of 3 feet and a cylinder diameter of 20 inches. A diagram similar to Fig. 315 is drawn, and it is found that the area of the portion CDE is .86 sq. in. The diagram was drawn so that a vertical height of one inch represents 40 pounds pressure per square inch, and a horizontal distance of one inch represents a crank-pin travel of $1\frac{1}{2}$ feet. Assume coefficient of unsteadiness E to be $\frac{1}{40}$, and the ratio n as 4. Find the necessary weight of the fly-wheel rim.

SOLUTION.—Piston speed $= \frac{2 \times 3 \times 100}{60} = 10$ ft. per sec. Average speed of crank-pin = speed of piston $\times \frac{\pi}{2} = 10 \times \frac{3.1416}{2} = 15.7$ ft. per sec. $= V_0$. Area of piston $= .7854 \times 20^2 = 314.16$ sq. in. Work per square inch of piston represented by area $CDE = .86 \times 40 \times 1\frac{1}{2} = 51.6$ ft. lb. $= H$. $W = \frac{AHg}{n^2 E V_0^2} = \frac{314.16 \times 51.6 \times 32.16}{4^2 \times \frac{1}{40} \times 15.7^2} = 5{,}288$ lb. $= 2.644$ tons. Ans.

In the above solution the weight of the hub and arms has been neglected, and the whole weight has been assumed as being concentrated in the rim. This is the usual engineering practice. The rim is made heavy enough to make the engine run within the required limits of steadiness, and the weight of the arms, hub, etc., whatever it may be, simply adds so much more to the steadiness of running.

REVERSING GEAR.

1329. In many engines, it is necessary that the ratio of expansion be often varied, and that the direction of working be often changed. An enormous number of mechanisms have been devised to accomplish this result. The earliest, as well as one of the most efficient, is the **Stephenson link motion,** shown in Fig. 316.

1330. Two eccentrics A and B are keyed to the shaft. The two eccentric rods E and F are fastened to a link L. The valve stem V is bolted to a block W, which fits into L.

When the mechanism is in the position shown in the figure, the eccentric rod E is in line with the valve stem, and the action is precisely similar to the action of an engine with a single eccentric. The eccentric A will govern the steam distribution, and B will simply have no effect whatever. The angular advance is AOC; consequently, the engine will run in the direction shown by the arrow.

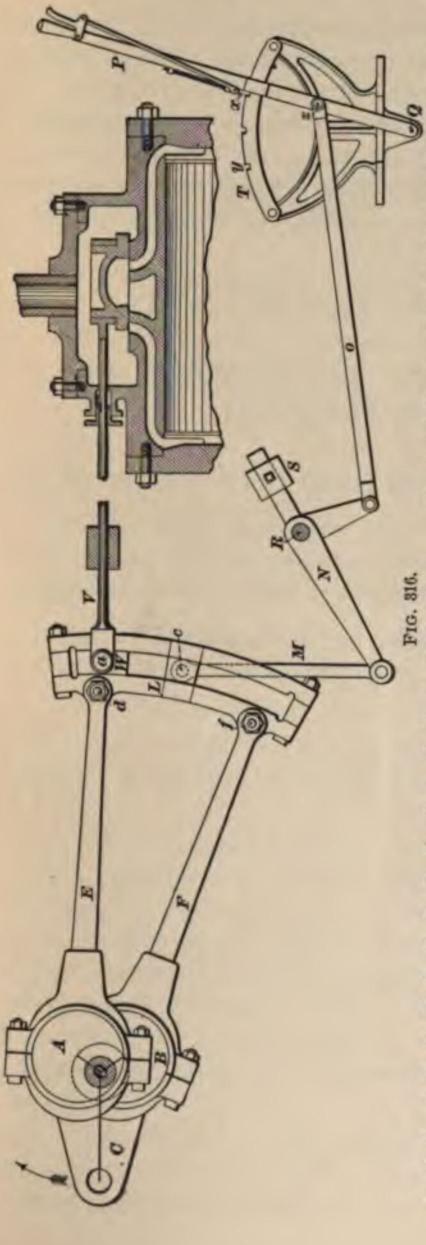


FIG. 316.

By means of the reversing lever *P*, and the links *O*, *N*, and *M*, pivoted at *R*, the link *L* can be raised until the eccentric rod *F* is in line with the valve stem *V*. The eccentric *B* will then impart motion to the valve; *B O C* will be the angular advance, and, since the eccentric must be ahead of the crank, the latter must move in the direction opposite to that shown by the arrow.

If the link be raised until the point of suspension *c* is just opposite the point *a* of the valve stem, the valve must partake equally of the motion of both eccentrics, and, hence, the engine will run in neither direction. If the link be raised so that *a* occupies a position somewhere between *c* and *d*, the engine will still run in the direction shown in the figure, but the motion of the point *a* will be influenced in some degree by the motion of *F*, and the travel of the valve will be less than when in the position of the figure. The result is an earlier cut-off and compression. In locomotive engines, the sector *T* has several notches. When starting, the lever is thrown into the last notch, thus giving the valve the longest possible travel and the engine the greatest power. After the train is started and the power required is less, the lever is thrown nearer the center, thus decreasing the travel of the valve, and, in consequence, the power developed by the engine.

PRINCIPLES
OF
REFRIGERATION.

FUNDAMENTAL PRINCIPLES OF REFRIGERATION.

THE MEANS OF PRODUCING REFRIGERATION.

1331. Refrigeration may be defined as the process of lowering the temperature of a body or of keeping the temperature below that of the atmosphere.

1332. Production of Cold.—Cold may be produced by one of the following processes:

1. A transfer of heat from a warmer to a colder body.

2. A chemical action, as exemplified by the so-called freezing mixtures. (See Instruction Paper, *Heat*, Art. **1142.**)

3. The adiabatic expansion of a gas. As explained in *Heat*, Art. **1164,** no heat is added to or abstracted from a gas during adiabatic expansion. If, therefore, the gas does work in pushing a piston, this work must be performed at the expense of the energy contained in the gas; the temperature of the gas will therefore fall, or, in other words, the gas will be cooled. It is to be carefully noted that if a gas does no work during adiabatic expansion, its temperature does *not* fall.

4. Evaporation of liquids having low boiling points. It was shown in *Heat*, Art. **1139,** that when a liquid is changed to a vapor, a certain quantity of heat, called the *latent heat of*

vaporization, must be added to the liquid to effect the change. hen this process of vaporization or evaporation takes place the presence of other bodies, the heat required for it is drawn from these bodies, and they are thereby cooled.

1333. The third method of producing cold mentioned in the preceding paragraph suggests a mechanical process of refrigeration. Referring to *Heat*, Art. **1178** and Fig. 232, let the volume of the gas in the cylinder be represented by the abscissa $O\,V_3$ and the pressure by the ordinate $V_3\,C$. Let the gas be compressed adiabatically until it has the volume represented by $O\,V_2$ and the pressure represented by $V_2\,B$. During this adiabatic compression no heat is added to or abstracted from the gas. A certain amount of work—represented by the area $B\,C\,V_3\,V_2$—is done on the gas by the piston and is stored up in the gas, thus causing a rise of temperature. When the gas is in the state represented by the point B, conceive the head a to be replaced by the conducting head b, and let the cylinder be placed in contact with a body which we will call the *hot body*. As the gas is compressed, heat passes freely from it to the hot body, the gas will remain at constant temperature, and the compression will be isothermal. Suppose the state of the gas at the end of compression to be represented by the point A; then the work done by the piston is represented by the area $B\,A\,V_1\,V_2$. Since the temperature of the gas does not change during this isothermal process, the gas neither receives nor loses energy, and the heat given up to the hot body must be the exact equivalent of the work done on the gas and represented by the area $B\,A\,V_1\,V_2$. Let the cylinder be removed from the hot body, and let the gas expand adiabatically from the state A to the state represented by D. Since the gas receives no heat during this expansion, the work done by it upon the piston—represented by the area $A\,D\,V_4\,V_1$—must be supplied by the gas itself; that is, the gas parts with enough of its energy to do this work, and as a result its temperature falls. We will assume that the expansion proceeds until the temperature of the gas is the same as at the initial state, represented by the point C. Suppose, now, that the cylinder is placed in contact with a second body, which we will designate the *cold body*, and that the head of the cylinder is again a perfect conductor. Let the gas now expand isothermally from the state D to the initial state C. During this expansion, the gas does the work represented by the area $D\,C\,V_3\,V_4$ upon the piston; but since the temperature of the gas remains constant, the energy stored up in it remains constant also; therefore the body in contact with the cylinder must give up to the gas the heat equivalent of the work done on the piston.

1334. It is obvious that an engine operating through the cycle just described is a refrigerating-machine. It takes a certain quantity of heat Q_2 from the cold body and adds a certain greater quantity of heat Q_1 to the hot body. The difference $Q_1 - Q_2$ represents the heat equivalent of the net work done by the piston upon the gas, and is represented by the area $A\,B\,C\,D$.

The refrigerating-machine, it will be observed, is a heat engine reversed. The latter goes through the Carnot cycle (see Fig. 232) in the order $A\,B\,C\,D$; the former in the order $A\,D\,C\,B$. The heat engine takes a quantity of heat Q_1 from the hot body, gives up a smaller quantity Q_2 to the cold body, and changes the difference $Q_1 - Q_2$ into mechanical work; the refrigerating-machine, on the contrary, takes the heat Q_2 from the cold body, changes the mechanical work into the heat $Q_1 - Q_2$, and delivers the heat Q_1 to the hot body.

1335. The Carnot cycle of a refrigerating-machine using a saturated vapor as a working fluid is shown in Fig. 317. Suppose the vapor to be in the state represented by the point A; that is, the volume is represented by $O\,V_1$ and the pressure by $V_1\,A$. The vapor is compressed adiabatically from the state A to the state B, and then isothermally from B to C. The pressure of any saturated vapor depends only upon the temperature; therefore, if the temperature remains constant, as in isothermal expansion, the pressure also remains constant, and the isothermal line $B\,C$ is a straight line parallel to the axis $O\,X$. (See *Steam and Steam Engines*, Art. **1210.**) During the compression from B to C, some or all of the vapor is condensed. The latent heat liberated by this change of state is absorbed by the hot body. The next stage of the process is adiabatic expansion from the state C to the state D, and this is followed by isothermal expansion from the state D to the original state A. During this isothermal expansion, the liquid changes back to a vapor, and the heat required to effect this vaporization is taken from the cold body with which the cylinder is in contact.

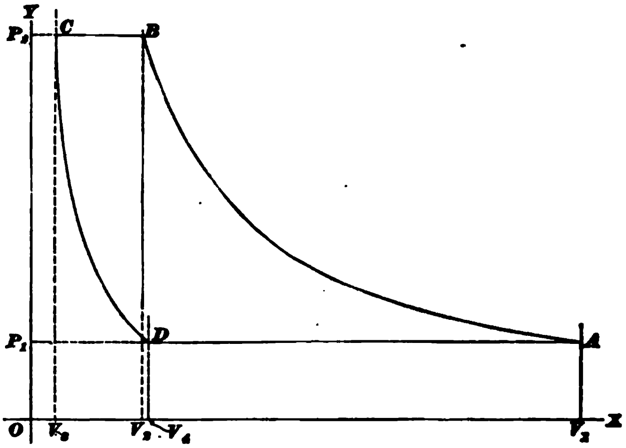


FIG. 817.

As in the previous case, a quantity of heat Q_1 is given up to the hot body, a smaller quantity Q_2 is abstracted from the cold body, and mechanical work equivalent to the difference $Q_1 - Q_2$ is done by the machine upon the working fluid. It is to be noted that in the present instance the refrigeration is produced by the vaporization of a volatile liquid.

1336. In actual operation, the cycle of the refrigerating-machine differs materially from that of the ideal Carnot cycle. In the ideal cases just described, the working fluid is assumed to remain in the cylinder. In practice this is not the case. The steam engine receives steam from the boiler, the source of heat, and rejects the exhaust steam into the condenser after extracting a certain portion of it in the form of work. Likewise the refrigerating-machine takes a supply of the working fluid from the cold body, does a certain amount of work upon it, rejects it at a higher temperature to the condenser, which we have heretofore called the hot body, and takes a new supply from the cold body.

In practice the cold body is either a room with non-conducting walls, in which are placed the articles to be chilled, or a non-freezing liquid called a **brine,** which after being cooled may be pumped through coils of pipes in the rooms to be cooled. The hot body is a condenser, through the coils of which the working fluid is forced by the compressor piston. Water is kept continually flowing over the coils of the condenser and carries away the heat generated by the isothermal compression BA, Fig. 232, or BC, Fig. 317.

1337. It is instructive to note the relation of the refrigeration process to the second law of thermodynamics. (See Art. **1177,** *Heat.*) This law asserts that *heat can not pass from a cold to a hot body by a self-acting process unaided by external agency.* In other words, whenever heat is transferred from a body to a warmer body, there must be some work expended to effect the transfer. In the process described above, the refrigerating-machine is the agent used in transferring the heat from the refrigerating room to the condensing water, and the work expended by the machine on the fluid is that required in the transfer. The transfer of heat may be expressed algebraically as follows:

Let Q_1 = heat delivered to condenser in B. T. U.;
Q_2 = heat taken from body cooled in B. T. U.;
W = work in foot-pounds done by engine on working fluid;
J = 778 = mechanical equivalent of heat.

Then, $$Q_1 = Q_2 + \frac{W}{J}. \qquad \textbf{(108.)}$$

CAPACITY AND EFFICIENCY OF REFRIGERATING-MACHINES.

1338. Refrigerating Capacity.—The capacity of a refrigerating-machine is a measure of its ability to abstract heat.

The unit of *refrigerating*, or *ice-melting*, *capacity* is the quantity of heat required to melt one ton (2,000 pounds) of ice at 32° F. to water at 32° F. Since 142.65 B. T. U. are required to melt one pound of ice at 32° F. (see Table 22), the unit of refrigerating capacity is equivalent to 142.65 B. T. U. × 2,000 = 285,300 B. T. U. The refrigerating capacity of a machine expressed in tons is given by the following formula:

Let H = B. T. U. abstracted from cold body in 24 hours;
h = B. T. U. abstracted from cold body in 1 hour;
F = refrigerating capacity expressed in tons.

Then,
$$F = \frac{H}{285{,}300}, \qquad (109.)$$
or
$$F = \frac{h}{11{,}887.5}. \qquad (110.)$$

ILLUSTRATION.—A refrigerating-machine abstracts 2,375,241 B. T. U. in 24 hours. The ice-melting capacity is

$$\frac{2{,}375{,}241}{285{,}300} = 8.33 \text{ tons.}$$

1339. Ice-Making Capacity.—The ice-making capacity is the number of tons of ice that a machine is capable of freezing in twenty-four hours. As the temperature of the water from which the machine freezes the ice varies from 50° to 95°, and as it is necessary to cool this water to 32° before any ice can be made, it will be seen that the ice-making capacity is variable and is largely affected by the conditions under which the machine operates. Owing to the necessity of cooling the water from which the ice is made from its initial temperature to a temperature below the freezing point, and owing to other losses, such as radiation, etc., the *ice-making* is only about 50 or 60 per cent. of the *ice-melting* capacity.

1340. Efficiency. — The theoretical maximum efficiency of a heat engine is given by the expression $\frac{T_1 - T_2}{T_1}$. (See *Heat*, Art. **1182.**) If we denote by Q_1 and Q_2, respectively, the heat given up by the hot body and the heat rejected to the cold body, it can readily be shown that this efficiency is also given by the ratio $\frac{Q_1 - Q_2}{Q_1}$. Referring to *Heat*, Art. **1182,** the heat equivalent of the work performed during isothermal expansion is $\left(\frac{2.3026\, c \log r_1}{778}\right) T_1$, and the heat equivalent of the work of isothermal compression is $\left(\frac{2.3026\, c \log r_2}{778}\right) T_2$. It is shown in Art. **1181** that $\frac{O\, V_2}{O\, V_1} = \frac{O\, V_3}{O\, V_4}$, or $r_1 = r_2$; hence, denoting the constant $\frac{2.3026\, c \log r_1}{778}$ by k, we have $Q_1 = k\, T_1$, $Q_2 = k\, T_2$, and $\frac{T_1 - T_2}{T_1} = \frac{k\, Q_1 - k\, Q_2}{k\, Q_1} = \frac{Q_1 - Q_2}{Q_1}$. This expression might have been written at once, for the efficiency of a heat engine is clearly the ratio of the heat transformed into mechanical work to the whole quantity of heat supplied.

In the case of the refrigerating-machine, the useful work is measured by the quantity of heat Q_2 removed from the cold body, and the work expended, that is, the work done by the machine upon the working fluid, expressed in heat-units, is $Q_1 - Q_2$ B. T. U. The ratio $\frac{Q_2}{Q_1 - Q_2}$ of the work obtained to the work expended is called the *efficiency* of the refrigerating-machine. If T_2 and T_1, respectively, denote the absolute temperatures at which the heat is abstracted from the cold body and delivered to the condenser, then, as in the case of the heat engine, $Q_2 = k\, T_2$ and $Q_1 = k\, T_1$; hence $\frac{Q_2}{Q_1 - Q_2} = \frac{T_2}{T_1 - T_2}$. Denoting the theoretical maximum efficiency by E, we have the following formulas:

$$E = \frac{Q_2}{Q_1 - Q_2}. \qquad (111.)$$

$$E = \frac{T_2}{T_1 - T_2}. \qquad (112.)$$

$$E = \frac{JQ_2}{W}. \qquad (113.)$$

In the last formula, J and W have the same significance as in Art. **1337.**

In the case of the steam engine, the efficiency increases with the range of temperature, and the effort of the engineer is to make the difference $T_1 - T_2$ as large as possible by increasing boiler pressure and lowering the temperature of the exhaust steam. With the refrigerating-machine, on the other hand, the expression $E = \frac{T_2}{T_1 - T_2}$ shows that the efficiency is increased by making the difference $T_1 - T_2$ smaller; that is, *the efficiency increases as the temperature of the cold body increases and the temperature of the condenser* (*hot body*) *decreases.* It is of course impossible to attain the maximum theoretical efficiency in practice on account of the losses of various kinds in the process, such as the friction of the machine, heat losses by radiation, etc. The work actually delivered to the compressor is always greater than the mechanical equivalent of the heat $Q_1 - Q_2$.

EXAMPLE.—If, per cubic foot of ammonia, 64.5 B. T. U. are carried from the cold room and 73.6 B. T. U. are delivered to the condenser, what is the theoretical maximum efficiency?

SOLUTION.— $E = \frac{Q_2}{Q_1 - Q_2} = \frac{64.5}{73.6 - 64.5} = \frac{64.5}{9.1} = 7.09.$ Ans.

1341. In general it is found that the efficiency as defined above is greater than 1, which is apparently a contradiction of the fundamental law that the efficiency of a machine is always less than 1. (See *Elementary Mechanics*, Arts. **949** and **950.**) This is due to the fact that in ordinary machines a certain amount of energy is delivered to the machine and a certain percentage of that energy appears as useful work, the remainder being expended in overcoming losses due to friction; thus the energy obtained is a part of that expended. In the refrigerating-machine, however, the energy obtained, that is, the heat abstracted from the cold body, is not a part of the energy supplied to the machine, and, in fact, has no direct connection with it. The work to be done by the compressor is *not* the mechanical equivalent of the heat abstracted, but represents only the difference between that heat and the heat delivered to the condenser. If, under exceptional circumstances, the cooling water used with the condenser should have a temperature lower than the temperature of the cold body, heat would of itself flow from the cold body to the condenser, and no compressor or working fluid would be required. This consideration shows that there is no necessary connection between the work of the compressor and the heat abstracted from the cold body, and as a consequence the ratio of one to the other may be either greater or less than 1.

1342. A refrigerating-machine is generally driven by a steam engine; therefore the energy delivered to the machine is contained primarily in the fuel fed to the furnace, usually coal. For this reason, it is customary in commercial work to measure the commercial efficiency or the *economy* of a refrigerating-machine by the pounds of ice-melting effect per pound of coal used. For every pound of coal consumed in the boiler to produce steam to operate the refrigerating-machine, a quantity of heat is abstracted from the cold body sufficient to melt a definite number of pounds of ice at 32° F. into water at 32° F. (See Art. **1338.**) This quantity of ice is a measure of the commercial efficiency of the machine.

EXAMPLE.—A refrigerating-machine having an actual capacity of 23.5 tons requires 4,350 pounds of coal per 24 hours to operate it. Required, the efficiency expressed in ice per pound of coal.

SOLUTION.— 23.5 tons = 47,000 lb.; 47,000 ÷ 4,350 = 10.8; hence, 10.8 pounds of ice are melted per pound of coal burned. Ans.

THE AIR REFRIGERATING-MACHINE.

1343. The air machine utilizes the fall of temperature that occurs when compressed air expands adiabatically and performs work. The principles underlying the operation of this machine are stated in Art. **1333.**

The general arrangement of an air refrigerating-machine is shown in Fig. 318. The machine consists essentially of a single-acting compression cylinder *A*, expansion cylinder *B*, also single-acting, a condenser *R*, and a cooler or refrigerator box *D*. The piston of the cylinder *A* is provided with suction valves *V*, *V* opening inwards, a discharge-valve *V'*, and also with a water-jacket *J*. The diameter of the cylinder *B* is slightly less than that of *A*. The piston is solid, but the cylinder-head is provided with two valves, an inlet valve *S* and an outlet valve *S'*, which are operated by the eccentrics *C* and *C'*. The pistons are driven by cranks set at 180°. The condenser *R* is a surface condenser and receives a current of cold water from the water-jacket *J* of the compression cylinder *A*. A receiver *R'* is connected with the condenser and also communicates with the inlet valve *S* of the expansion cylinder *B*.

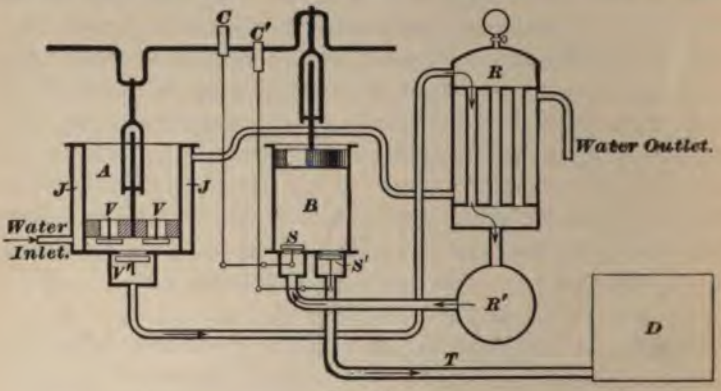


FIG. 318.

The air at ordinary pressure is taken into the cylinder *A* through the valves *V*, *V*, and is compressed adiabatically until the pressure becomes sufficient to open the valve V'. The air then passes into the condenser R, where it comes in contact with the cold surfaces of that vessel. The adiabatic compression has raised the temperature of the air, but in passing through the condenser, some of the heat contained in the air is given up to the cold water circulating through the condenser, and the temperature is lowered nearly to that of the surrounding air. During this time the valve S of the expansion cylinder B opens and permits an amount of air equal in weight to that expelled from A to pass from the receiver R' into the cylinder. The valve S closes and the air in the cylinder B expands, forcing the piston forwards and doing a certain amount of work, which may be deducted from the work of the compression. This expansion of the air in the cylinder B and the performance of work in forcing the piston forwards is at the expense of the energy stored in the air. The air therefore gives up sufficient heat to do the mechanical work, and as a result its temperature falls. As the air on entering B was at a normal temperature, the expansion brings the temperature below that of the surrounding objects. In other words, the air is cooled.

When the piston in B reaches the upper limit of its stroke, the valve S' opens, and as the piston descends, the cooled air escapes by means of the pipe T into the refrigerator box D.

The difference between the work done on the air in the compression cylinder and that done by the air in the expansion cylinder, and, in addition, the work required to overcome the friction of the entire machine, must be supplied by a steam engine or other motor.

1344. The operation of the actual air machine as just outlined differs in some degree from the action of the ideal machine as described in Art. **1333,** and the cycle of the actual machine differs from the Carnot cycle. The cycle has, as usual, four operations; they are: compression, cooling, expansion, and refrigeration. Let $O\ V_2$, Fig. 319, represent the volume of air taken into the compression cylinder per stroke. During compression the volume decreases and the pressure rises, as indicated by the curve *A B*. The water circulating through the jacket carries away some of the heat developed, so that the compression is not strictly adiabatic, and the temperature at the end of compression is less than it would be if the curve *A B* were an adiabatic.

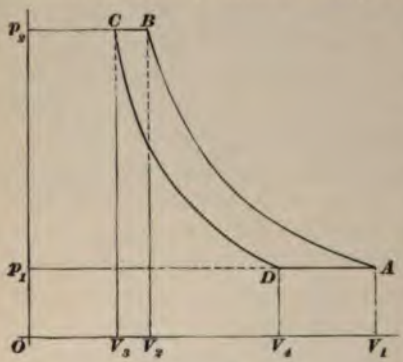


FIG. 319.

As the air from *A*, Fig. 318, is pushed into the condenser *R*, an equal weight is delivered to the cylinder *B*; consequently the weight of air in *R*, *R'* and the conducting pipes is practically constant, and the pressure remains practically constant also. During the second operation, therefore, the air is cooled in the condenser at constant pressure. According to Gay-Lussac's law (*Pneumatics*, Art. **1054**), the cooling is accompanied by a decrease in volume. If $O\,V_2$, Fig. 319, represents the volume before cooling, $O\,V_3$ will represent the volume after cooling; and the operation will be represented by the constant-pressure line *B C*. The temperature of the air in the state represented by *C* is not much above that of the surrounding atmosphere.

The air now enters the cylinder *B* and expands adiabatically; the temperature, which was normal at the beginning of the expansion, must fall much below the temperature of surrounding objects; the volume increases from $O\,V_3$ to $O\,V_4$. The cylinder *B* is made of such diameter that when the piston reaches the end of its stroke, the pressure of the air will be just that of the atmosphere. This third operation is represented by the curve *C D*.

The air is now pushed into the cooler *D* by the return stroke of the piston. Being colder than the surrounding objects, it absorbs heat from them and expands at the constant pressure $V_4 D$. If the supply to the compressor had been of the same temperature as the cooler; for example, if the compressor had drawn its supply from the cooler, the volume in the cooler would increase from $O V_4$ to the original volume $O V_1$, as indicated by the line $D A$, and the cycle would be closed. Usually, however, it is more convenient to reject the air into the cooler D and draw the fresh supply from the atmosphere, which has a much higher temperature. In this case, it is evident that the air does not return to its original state in the cooler, and the cycle is not closed.

1345. General Theory.—In the following discussion, it will be assumed, for the sake of simplicity, that compression and expansion are adiabatic and that the air is drawn into the compressor from the cooling chamber, so that the machine works through a closed cycle. The clearance of the two cylinders will be neglected. Referring to the diagram, Fig. 319, let V_a, V_b, V_c, V_d denote the volumes, T_a, T_b, T_c, T_d the absolute temperatures, and t_a, t_b, t_c, t_d the ordinary temperatures of the air when in the states represented by the points A, B, C, and D, respectively; and let p_1 and p_2 denote the pressures of the air during the operations $D A$ and $B C$, respectively; also let s_p and s_v denote, respectively, the specific heat of air at constant pressure and constant volume, and let M denote the weight of air used per stroke of the compressor.

The temperatures t_a of the cooling chamber and t_c of the air as it leaves the condenser are known or assumed, and the temperatures t_b and t_d can be obtained by formula **86,** *Heat.* Thus,

$$\frac{T_b}{T_a} = \left(\frac{p_2}{p_1}\right)^{.29078} \text{ and } \frac{T_d}{T_c} = \left(\frac{p_1}{p_2}\right)^{.29078} = \frac{T_a}{T_b}. \quad (a)$$

To find the volume of the air at the end of compression, we have, from formula **81,**

$$p_1 V_a^{1.41} = p_2 V_b^{1.41},$$

or

$$V_b = V_a \left(\frac{p_1}{p_2}\right)^{\frac{1}{1.41}} \quad (b)$$

The volume of the air during the cooling in the condenser decreases from V_b to V_c at constant pressure. According to formula 71,

$$p_2 V_b = R M T_b,$$

and $$p_2 V_c = R M T_c.$$

Dividing, $$\frac{V_b}{V_c} = \frac{T_b}{T_c},$$

or $$V_c = V_b \frac{T_c}{T_b}. \quad (c)$$

Finally, $$p_2 V_c^{1.41} = p_1 V_d^{1.41}.$$

or $$V_d = V_c \left(\frac{p_2}{p_1}\right)^{\frac{1}{1.41}}.$$

Since from (*b*) $$\left(\frac{p_1}{p_2}\right)^{\frac{1}{1.41}} = \frac{V_b}{V_a},$$

we have $$V_d = V_c \frac{V_a}{V_b}, \text{ or } \frac{V_d}{V_c} = \frac{V_a}{V_b}. \quad (d)$$

The preceding formulas enable us from the assumed data to calculate the volume and temperature of the gas at each of the four points *A*, *B*, *C*, and *D*.

The heat given up by the air to the condenser is

$$Q_1 = s_p M (t_b - t_c). \quad (e) \quad \text{(See Art. } \mathbf{1135}, \textit{Heat.})$$

The heat absorbed by the air from the cooler as it expands from *D* to *A* is

$$Q_2 = s_p M (t_a - t_d). \quad (f)$$

The specific heat s_p is used for the reason that the air passes through the condenser and through the cooler at constant pressure, as shown by the lines *B C* and *D A*.

The heat equivalent of the work done on the air—represented by the area *A B C D*—must be precisely the difference between the heat delivered to the condenser and that abstracted from the cooler. Hence, denoting the net work by *W*, we have

$$\frac{W}{J} = Q_1 - Q_2 = s_p M[(t_b - t_c) - (t_a - t_d)],$$

or

$$W = J s_p M[(t_b - t_c) - (t_a - t_d)]$$

$$= J s_p M[(T_b - T_c) - (T_a - T_d)]. \quad (g)$$

The theoretical efficiency is therefore

$$E = \frac{J Q_2}{W} = \frac{J s_p M(T_a - T_d)}{J s_p M[(T_b - T_c) - (T_a - T_d)]}$$

$$= \frac{T_a - T_d}{(T_b - T_c) - (T_a - T_d)}. \quad (h)$$

Since $\frac{T_d}{T_c} = \frac{T_a}{T_b}$, it can readily be shown that the expression reduces to

$$E = \frac{T_a}{T_b - T_a} = \frac{T_d}{T_c - T_d}. \quad (i)$$

Since the net work per stroke is W, if we denote the number of strokes per minute by n, the horsepower required to drive the machine is

$$H = \frac{n W}{33{,}000}. \quad (j)$$

The gross horsepower required will of course be much greater, on account of the friction of the two pistons and of the other parts of the machine.

If desired, the work W may be expressed in terms of the pressures and volumes instead of temperatures. Thus, using formula **88**, *Heat*,

$$W_{ab} (\text{area } A\,B\,V_b\,V_a) = \frac{144\,(p_2 V_b - p_1 V_a)}{.41}.$$

$$W_{bc} (\text{area } B\,C\,V_c\,V_b) = 144\,p_2\,(V_b - V_c).$$

$$W_{cd} (\text{area } C\,D\,V_d\,V_c) = \frac{144\,(p_2 V_c - p_1 V_d)}{.41}.$$

$$W_{da} (\text{area } D\,A\,V_a\,V_d) = 144\,p_1\,(V_a - V_d).$$

$$W = W_{ab} + W_{bc} - W_{cd} - W_{da} =$$

$$144 \times \frac{1.41}{.41}\,[p_2(V_b - V_c) - p_1(V_a - V_d)]. \quad (k)$$

The factor 144 is used to reduce the pressures from pounds per square inch to pounds per square foot.

The heat Q_1 is given up by the air in the condenser. If G denote the weight of cooling water used per stroke, and t_e and t_f the temperature of the water on entering and leaving the condenser, the heat absorbed by the water per stroke is $G(t_f - t_e)$ B. T. U.

Hence, $$G(t_f - t_e) = Q_1 = s_p M(t_b - t_c),$$

or $$G = \frac{s_p M(t_b - t_c)}{t_f - t_e}. \quad (l)$$

The capacity of the compression cylinder (neglecting clearance) is V_a and that of the expansion cylinder is V_d. To find the ratio of these volumes, we have

$$p_1 V_d = RMT_d,$$

and $$p_1 V_a = RMT_a;$$

whence, dividing one equation by the other,

$$\frac{V_d}{V_a} = \frac{T_d}{T_a}. \quad (m)$$

1346. To illustrate the application of the equations developed in the preceding article, the horsepower and approximate cylinder dimensions for an air refrigerating-machine will be calculated from the following data: The machine is required to have an ice-melting capacity of 300 pounds of ice per hour. The pressure (absolute) in the cold chamber is 14.7 pounds per square inch, and the temperature is 40° F. The pressure in the expansion cylinder at cut-off is 50 pounds gauge, or 64.7 pounds absolute. The initial and final temperatures of the cooling water are 65° and 85°, and the temperature of the air as it leaves the cooler is 95° F. The machine is single-acting and makes 75 strokes per minute.

The heat Q_2 abstracted per hour is

$$142.65 \text{ B. T. U.} \times 300 = 42{,}795 \text{ B. T. U.}$$

The heat abstracted per stroke is $\frac{42{,}795}{75 \times 60} = 9.51$ B. T. U.

The absolute temperature at the end of compression is

$$T_b = T_a\left(\frac{p_2}{p_1}\right)^{.29078} = (40° + 460°) \times \left(\frac{64.7}{14.7}\right)^{.29078} = 769.33°;$$

hence, $t_b = 769.33° - 460° = 309.33°$ F.

The temperature $T_c = 95° + 466° = 555°$; according to formula (*a*),

$$T_d = T_c\frac{T_a}{T_b} = 555 \times \frac{500}{769.33} = 360.703°;$$

therefore, $t_d = 360.703° - 460° = -99.297°$.

Substituting known values in formula (*f*),

$$Q_2 = s_p M(t_a - t_d),$$

$$9.51 = .23751 \times M \times [40 - (-99.297)],$$

or $$M = \frac{9.51}{.23751 \times (40 + 99.297)} = .2874 \text{ pound.}$$

The volume V_a, which is the capacity of the compressor cylinder, can now be found from formula **71**, *Heat*. The weight of air per stroke is .2874 pound, and the absolute temperature T_a at which the air is admitted to the compressor is 40° + 460° = 500°; hence the volume is

$$V_a = \frac{RMT_a}{p_1} = \frac{.37052 \times .2874 \times 500}{14.7} = 3.622 \text{ cu. ft.}$$

For the volume V_d of the expansion cylinder, we have, from formula (*m*),

$$V_d = V_a\frac{T_d}{T_a} = 3.622 \times \frac{360.703}{500} = 2.613 \text{ cu. ft.}$$

These volumes should be increased about 20 per cent. to allow for the loss due to clearance and to fall of pressure caused by the resistance of valves and passages, and for various imperfections in the operation of the machine. Making this allowance, the cylinders may have the following dimensions:

Diameter of compression cylinder,	22 inches ;
Diameter of expansion cylinder,	18½ inches ;
Length of stroke of both cylinders,	20 inches.

The work per stroke is

$$W = J\,s_p\,M[(T_b - T_c) - (T_a - T_d)] =$$
$$778 \times .23751 \times .2874 \times [(769.33 - 555) - (500 - 360.703)] =$$
$$3{,}984.7 \text{ foot-pounds.}$$

The net horsepower required is

$$H = \frac{n\,W}{33{,}000} = \frac{75 \times 3{,}984.7}{33{,}000} = 9.06 \text{ H. P.}$$

The gross horsepower will be about $\frac{4}{3}$ of the net horsepower, or 12.08 H. P.

The cooling water required per minute is

$$75\,G = 75\,\frac{s_p\,M\,(t_b - t_c)}{t_f - t_e} =$$
$$\frac{75 \times .23751 \times .2874 \times (309.33 - 95)}{85 - 65} = 54.86 \text{ pounds.}$$

The theoretical efficiency of the machine is

$$E = \frac{T_a}{T_b - T_a} = \frac{500}{769.33 - 500} = 1.856.$$

Since the machine transfers heat from the cold room, where the temperature is 40°, or 500° absolute, to the condensing water, the final temperature of which is $T_f = 85° + 460° = 545°$, the maximum possible efficiency of a perfect engine working between these temperatures through a Carnot cycle is

$$E_m = \frac{T_a}{T_f - T_a} = \frac{500}{545 - 500} = 11.11. \quad \text{(See Art. \textbf{1340}.)}$$

The theoretical efficiency is therefore only about $\frac{1}{6}$ of the maximum theoretical efficiency.

1347. Capacity of Air Machine.—A formula for the theoretical ice-melting capacity of an air refrigerating-machine may be derived as follows: Let T_1 denote the temperature of the air entering the compressor; if the cycle is closed, T_1 is the same as T_a, Art. **1345,** but in general this is not the case. As in the previous discussion, V_a and V_d are the volumes of the compression and expansion cylinders,

T_d the temperature of the air at the end of expansion, and T_a the temperature of the cooling chamber. Let p_1 denote the pressure at the end of expansion; this pressure should be that of the atmosphere, 14.7 pounds, if the expansion cylinder is properly proportioned and the valves are set correctly.

The weight of air M entering the compressor per stroke is given by the formula

$$p_1 V_a = RMT_1,$$

whence

$$M = \frac{p_1 V_a}{R T_1}.$$

Provided there is no loss of air in passing through the machine, the same weight of air must be delivered by the expansion cylinder; that is,

$$M = \frac{p_1 V_d}{R T_d}.$$

The heat abstracted per stroke is

$$Q_2 = s_p M (T_a - T_d) = \frac{s_p p_1 V_a}{R T_1} (T_a - T_d).$$

Using the second value of M,

$$Q_2 = \frac{s_p p_1 V_d}{R T_d} (T_a - T_d) = \frac{s_p p_1 V_d}{R} \left(\frac{T_a}{T_d} - 1 \right).$$

If n denotes the number of strokes per minute, the heat H extracted in 24 hours is given by the expression

$$H = 24 \times 60 \times \frac{n s_p p_1 V_a}{R T_1} (T_a - T_d) =$$

$$24 \times 60 \times \frac{n s_p p_1 V_d}{R} \left(\frac{T_a}{T_d} - 1 \right).$$

Now, making use of formula **109**, the ice-melting capacity is

$$F = \frac{H}{285,300} = \frac{24 \times 60}{285,300} \times \frac{n s_p p_1 V_a}{R T_1} (T_a - T_d) =$$

$$\frac{24 \times 60}{285,300} \times \frac{n s_p p_1 V_d}{R} \left(\frac{T_a}{T_d} - 1 \right).$$

Substituting the numerical values .23751 and .37052 for s_p and R, we obtain finally

$$\left.\begin{aligned} F &= .003235\,n\,p_1\frac{V_a}{T_1}(T_a - T_d), \\ \text{or}\quad F &= .003235\,n\,p_1 V_d\left(\frac{T_a}{T_d} - 1\right). \end{aligned}\right\} \qquad \textbf{(114.)}$$

If the weight of air is found from the delivery of the expansion cylinder, the pressure p_1 may be found by means of an indicator. If an indicator is not available, the pressure may be taken as 14.7 lb. per square inch. The temperature T_d and T_a must also be determined.

EXAMPLE.—The diameter of the expansion cylinder is 21 inches and the stroke is 24 inches. The temperature of the air after expansion is −52°, the temperature of the refrigerating chamber is 33°, and the pressure in the chamber is 14.7 lb. per sq. in. The machine makes 70 strokes per minute. Required, the theoretical ice-melting capacity.

SOLUTION.—The volume V_d of the cylinder is $\frac{.7854 \times 21^2 \times 24}{1,728} =$ 4.81 cu. ft.

$$T_a = 460° + 33° = 493°, \text{ and } T_d = 460° - 52° = 408°.$$

Substituting these values in formula **114**,

$$F = .003235 \times 70 \times 14.7 \times 4.81 \times \left(\frac{493}{408} - 1\right) = 3.335,$$

the ice-melting effect in tons per 24 hours. Ans.

1348. Economy of the Air Machine.—The ideal air refrigerating-machine, like the ideal hot-air engine, has theoretically a high efficiency. In practice, however, the air machine has proved to be a wasteful and uneconomical machine, and it is at present rarely used, save under exceptional circumstances. The principal reasons for the inefficiency of the actual air machine are as follows:

1. *Friction.*—As the cooling effect is accomplished by means of the increase of the sensible heat in the air and not by the latent heat of evaporation, as in the case of the ammonia machine, it follows that the compression cylinder

must be large. In practice, the cylinder of the air machine has about twenty times the capacity of that of an ammonia machine of the same tonnage; the usual quantity of air delivered per minute is about *one hundred cubic feet per ton of ice-melting capacity.* This increase in the size of the cylinders as compared with those of other classes of machines means a much greater expenditure of work in friction. In actual practice, this friction loss amounts to about 25 per cent. of the total work.

2. *Clearance.*—In the case of the air machine there are two cylinders, a compression and an expansion cylinder, instead of the single compression cylinder of the ammonia machine. There is, therefore, at least double the clearance space. The air machines are usually built with a short stroke, which also increases the clearance per cubic foot of air pumped.

Clearance spaces greatly decrease the efficiency of any machine that has to compress a gas, for if the piston does not come up close to the head at the end of its stroke, it can not expel all the gas or air, and the portion that is left expands again in the cylinder and virtually cuts down the capacity of the machine by that amount. It has been, therefore, the study of all builders of machines for compressing gases to do away with clearance spaces as much as possible.

3. *Cylinder Superheating.*—When air is compressed, it becomes heated and naturally heats the walls of the cylinder. The cylinder being hot, the air drawn in during the next stroke also becomes heated. This heating reduces the density of the air, which affects both capacity and economy. In order to overcome this difficulty, the cylinder was jacketed, but this only partly remedied the defect, owing to the slow transmission of heat through the walls of the cylinder. The Bell-Coleman Co. then added to their machine water injection during compression. The injection did away largely with the cylinder superheating, but augmented another defect, namely:

4. *Moisture.*—Air at any ordinary temperature can hold a certain amount of water vapor in suspension. The limit, or point of saturation, that is, the point at which the air can hold no more water vapor, is called the dew point. When this point is reached, the excess of moisture above that which the air is able to hold is precipitated in the form of dew. The weight of moisture contained in a given volume of air at the dew point is not the same for all temperatures; in fact, air will hold in suspension four times the weight of moisture at 72° that it will at 32°. Assume that the air on entering the expansion cylinder is at a temperature of 72° and is saturated with moisture. As the temperature falls during expansion, the water is gradually precipitated out and condenses on the walls of the cylinder. The water cools as the expansion goes on until it reaches 32°, and then it freezes. The condensation, cooling, and freezing of the water take a great deal away from the useful effect of the machine. Besides, the snow, which is the result of freezing the moisture, often gives trouble by clogging the valves.

The Haslam Foundry and Engineering Co., Derby, England, make what is known as a "dry-air" system. They

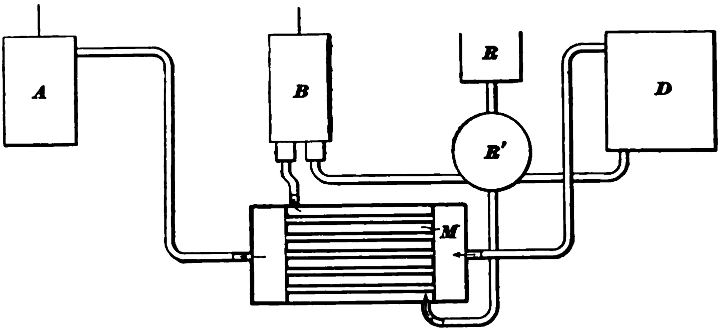


FIG. 320.

place a *drier* in the suction-pipe from the condenser to the expansion cylinder. The compression cylinder *A*, Fig. 320, takes the cold air from the refrigerator box *D*. On its way to *A*, this cold air passes through the pipes of the drier *M*. The cold strikes through these pipes and cools the air surrounding the pipes on its way from the receiver *R'* to the expansion cylinder *B*. The air gives up a large percentage of its moisture in the drier, and the frosting in the cylinder *B* is much diminished.

LATENT-HEAT REFRIGERATING-MACHINES.

FLUIDS USED AS REFRIGERATING AGENTS.

1349. The air refrigerating-machine produces its refrigerating effect by means of the fall of temperature incident to adiabatic expansion. In all other refrigerating-machines, the abstraction of heat is brought about by the vaporization of some liquid having a low boiling point. Such machines may be classed as **latent-heat** refrigerating-machines.

1350. Theoretically, any volatile liquid may be used as a working fluid in a latent-heat machine; there are, however, various considerations of a practical nature that govern the choice of the liquid. The chief requisites of the fluid used are: (1) It should vaporize at a low temperature when at ordinary atmospheric pressure. (2) It should have a high latent heat.

The fluids that have been used in compression machines are ether, sulphur dioxide, anhydrous ammonia, and Pictet's fluid.

1351. The following table gives the boiling points and latent heats of various substances at atmospheric pressure, 14.7 pounds per square inch.

TABLE 23.

Substance.	Temperature of Boiling Point.	Latent Heat, B. T. U.	Specific Heat of Liquid.
Nitric Acid...........	248° F.	....	....
Saturated Brine	226° F.	....	....
Water................	212° F.	966	1.0000
Alcohol...............	173° F.	....	....
Chloroform...........	140° F.	....	....
Ether, Sulphurous	95° F.	170	.5299
Ether, Methyl	– 10° F.	....	....
Sulphur Dioxide	14° F.	168.7	.4100
Anhydrous Ammonia..	– 28.5° F.	573	1.0058
Carbon Dioxide........	– 140° F.	141	.9550

Under a pressure of 342 pounds per square inch, carbon dioxide boils at a temperature of 5°. Its latent heat under the same conditions is 121.5.

ETHER.

1352. Early in the history of ice-making and refrigerating machines, ether was almost universally used as the working fluid. This was due to its high condensing temperature and consequent low condensing pressure. This low condensing pressure made it possible to use compression pumps of ordinary construction, very much after the style and pattern of air-pumps. However, the disadvantages in the use of ether were found to be very great; the first cost of ether is considerable; but the greatest objection to it is its inflammability and its liability to explode when mixed with air.

SULPHUR DIOXIDE.

1353. The objections to ether led to further investigation. Sulphur dioxide was found to be more efficient than ether, for, though it required a higher condensing pressure, it did not require to be evaporated under a vacuum. Consequently the compression pumps were made somewhat smaller for a given capacity, but were built stronger and more attention was given to the elimination of clearance spaces. The temperatures produced with sulphur dioxide, though lower than that obtained with ether, were not sufficiently low.

Table 24 gives the leading properties of sulphur dioxide.

TABLE 24.

PROPERTIES OF SATURATED SULPHUR DIOXIDE.

Temperature of Ebullition in Deg. F.	Absolute Pressure in Lb. per Sq. In.	Total Heat Reckoned from 32° Fahr.	Heat of Liquid Reckoned from 32° Fahr.	Latent Heat of Vaporization.	Density of Vapor or Weight of 1 Cubic Ft.
Deg. F.	Lb.	B. T. U.	B. T. U.	B. T. U.	Lb.
−40	3.16	155.22	−17.76	172.98	.048
−31	4.23	156.39	−16.55	172.94	.062
−22	5.56	157.55	−15.05	172.60	.079
−13	7.23	158.69	−13.26	171.95	.099
− 4	9.27	159.82	−11.18	171.00	.124
5	11.76	160.93	− 8.82	169.75	.154
14	14.75	162.02	− 6.17	168.19	.190
23	18.31	163.10	− 3.23	166.33	.232
32	22.53	164.16	0.00	164.16	.282
41	27.48	165.21	3.52	161.69	.341
50	33.26	166.24	7.32	158.92	.410
59	39.93	167.25	11.41	155.84	.491
68	47.62	168.25	15.79	152.46	.584
77	56.39	169.23	20.45	148.78	.692
86	66 37	170.20	25.41	144.79	.819
95	77.64	171.15	30.65	140.50	.965
104	90.32	172.08	36.18	135.90	1.131

PICTET FLUID.

1354. It was found by Prof. Pictet, a Swiss physicist, that a mixture of 97% of sulphur dioxide and 3% carbon dioxide, commonly known as carbonic acid gas, gives a boiling point 14° F. lower than pure sulphur dioxide. This liquid, or rather mixture, has been since known as **Pictet Fluid.** Its latent heat has never been closely determined, but is very nearly the same as that of pure sulphur dioxide.

CARBON DIOXIDE.

1355. This liquid has the lowest boiling point of any of the fluids employed in refrigeration. Under a gauge pressure of 200 pounds per square inch, it will have a

TABLE 25.

PROPERTIES OF SATURATED CARBON DIOXIDE.

Temperature of Ebullition in Deg. F.	Absolute Pressure in Lb. Per Sq. In.	Total Heat from 32° F.	Heat of Liquid from 32° F.	Latent Heat of Vaporization.	Density of Vapor, or Weight of 1 Cu. Ft.
– 22	210	98.35	– 37.80	136.15	2.321
– 13	249	99.14	– 32.51	131.65	2.759
– 4	292	99.88	– 26.91	126.79	3.265
5	342	100.58	– 20.92	121.50	3.853
14	396	101.21	– 14.49	115.70	4.535
23	457	101.81	– 7.56	109.37	5.331
32	525	102.35	0.60	102.35	6.265
41	599	102.84	8.32	94.52	7.374
50	680	103.24	17.60	85.64	8.708
59	768	103.59	28.22	75.37	10.356
68	864	103.84	40.86	62.98	12.480
77	968	103.95	57.06	46.89	15.475
86	1,080	103.72	84.44	19.28	21.519

temperature of about − 22° F. Its condensing pressure is correspondingly high, being about 900 pounds per square inch for a water temperature of 70° F.

Table 25 gives the properties of carbon dioxide at different temperatures.

AMMONIA.

1356. Chemical Composition.—One atom of nitrogen combines with three atoms of hydrogen to form one molecule of **ammonia**; this is the only combination of these two elements. The ordinary ammonia of commerce is a **solution** of ammonia gas in water, and is properly known as **aqua ammonia.** The gas which passes off from the aqua ammonia is the ammonia formed by the combination of nitrogen and hydrogen. When this gas is **entirely free** from vapor of water, it is called **anhydrous** ammonia gas.

1357. Physical Properties.—Ammonia gas, when liquefied under a high pressure and allowed to evaporate under atmospheric pressure, gives a temperature of 28.5° F. below zero. Liquid anhydrous ammonia when subjected to a temperature of − 115° F. freezes and forms a solid. In this state it is almost odorless and is heavier than the liquid.

Ammonia has no effect on either iron or steel, but rapidly corrodes copper and brass. It is therefore necessary to make the parts of ammonia machines out of the former metals.

At a temperature of 900° F., the gas is resolved into its constituent elements. But it is probable that this dissociation occurs, to a limited degree, at much lower temperatures.

Ammonia is not inflammable at ordinary temperatures, but if mixed with oxygen will burn with a pale-yellow flame. The liquid will not explode, but when run into drums or flasks, room should be left for expansion. Like almost all liquids, ammonia expands when heated, and if sufficient space is not left for expansion, the flask is likely to burst if exposed to a high temperature.

TABLE 26.

PROPERTIES OF SATURATED AMMONIA.

TEMPERATURE.	PRESSURE, ABSOLUTE.	Heat of Vaporization, Thermal Units.	Volume of Vapor per Lb., Cu. Ft.	Volume of Liquid per Lb., Cu. Ft.	Weight of a Cu. Ft. of Vapor, Pounds.
Degrees F.	Lb. per Sq. In.				
t	p	r	v	v_1	w
− 40	10.69	579.67	24.3700	.0234	.0410
− 35	12.31	576.69	21.2900	.0236	.0467
− 30	14.13	573.69	18.6600	.0237	.0535
− 25	16.17	570.68	16.4100	.0238	.0609
− 20	18.45	567.67	14.4800	.0240	.0690
− 15	20.99	564.64	12.8100	.0242	.0779
− 10	23.77	561.61	11.3600	.0243	.0878
− 5	26.93	558.56	10.1200	.0244	.0988
0	30.37	555.50	9.0400	.0246	.1109
+ 5	34.17	552.43	8.0600	.0247	.1241
+ 10	38.55	549.35	7.2300	.0249	.1384
+ 15	42.93	546.26	6.4900	.0250	.1540
+ 20	47.95	543.15	5.8400	.0252	.1712
+ 25	53.43	540.03	5.2600	.0253	.1901
+ 30	59.41	536.92	4.7500	.0254	.2105
+ 35	65.93	533.78	4.3100	.0256	.2320
+ 40	73.00	530.63	3.9100	.0257	.2588
+ 45	80.66	527.47	3.5600	.0260	.2809
+ 50	88.96	524.30	3.2500	.0260	.3076
+ 55	97.93	521.12	2.9600	.0260	.3378
+ 60	107.60	517.93	2.7000	.0265	.3704
+ 65	118.03	514.73	2.4800	.0266	.4034
+ 70	129.21	511.52	2.2700	.0268	.4405
+ 75	141.25	508.29	2.0800	.0270	.4808
+ 80	154.11	504.66	1.9100	.0272	.5236
+ 85	167.86	501.81	1.7700	.0273	.5649
+ 90	182.80	498.11	1.6400	.0274	.6098
+ 95	198.37	495.29	1.5100	.0277	.6622
+ 100	215.14	491.50	1.3900	.0279	.7194
+ 105	232.98	488.72	1.2890	.0281	.7757
+ 110	251.97	485.42	1.2030	.0283	.8312
+ 115	272.14	482.41	1.1210	.0285	.8912
+ 120	293.49	478.79	1.0410	.0287	.9608
+ 125	316.16	475.45	.9699	.0289	1.0310
+ 130	340.42	472.11	.9051	.0291	1.1048
+ 135	365.16	468.75	.8457	.0293	1.1824
+ 140	392.22	465.39	.7910	.0295	1.2642
+ 145	420.49	462.01	.7408	.0297	1.3497
+ 150	450.20	458.62	.6946	.0299	1.4396
+ 155	481.54	455.22	.6511	.0302	1.5358
+ 160	514.40	451.81	.6128	.0304	1.6318
+ 165	549.04	448.39	.5765	.0306	1.7344

1358. The leading properties of ammonia that are dependent upon the pressure or temperature are given in Table 26, which is taken from Wood's "Thermodynamics." The table is calculated from the following formulas, which are based partly on experimental data and partly on thermodynamic principles :

Let p = absolute pressure of gas or vapor in pounds per square inch;
t = temperature of vapor, Fahrenheit;
v = volume of one pound of vapor;
v_1 = volume of one pound of ammonia liquid;
w = weight of one cubic foot of vapor;
r = latent heat of vaporization in B. T. U.

$$\text{Then, } \log p = 6.2495 - \frac{2,196}{460.66 + t}. \qquad \textbf{(115.)}$$

$$v_1 = \frac{.016}{.6502 - .000778\,t}. \qquad \textbf{(116.)}$$

$$v = v_1 + .00107\frac{r}{p}t + .4923\frac{r}{p}. \qquad \textbf{(117.)}$$

$$w = \frac{1}{v}.$$

$$r = 555.5 - .613\,t - .000219\,t^2. \qquad \textbf{(118.)}$$

It will be noted that with ammonia vapor, as with all other saturated vapors, the temperature depends only upon the pressure, and *vice versa;* the relation between the temperature and pressure is given by formula **115.**

A comparison of the properties of ammonia with those of other refrigerating fluids shows that the latent heat of vaporization of ammonia is much greater than that of other fluids. This property makes ammonia especially desirable as a refrigerating agent, because, on account of the high latent heat, a greater refrigerating effect per pound of fluid circulated can be obtained with ammonia than with the other agents.

1359. Specific Heat.—The specific heat of liquid anhydrous ammonia has been variously computed to be from 1 to 1.2 that of water. The latest determinations show it to be 1.0058, virtually the same as water. The specific heat of ammonia gas is .508.

1360. Saturated and Superheated Gas.—When the temperature of the ammonia gas is the same as that of the boiling point of liquid anhydrous ammonia due to the pressure of the gas, the gas, or rather vapor, is at its greatest density and is said to be saturated. If heat is applied to this saturated gas so that its temperature is increased while the pressure remains constant, the gas becomes superheated and approaches very nearly a perfect gas in its properties. (See *Steam and Steam Engines*, Arts. **1191** and **1192.**)

The relation between the pressure volume and temperature of superheated ammonia gas has not yet been accurately determined. For ordinary purposes, however, the following equation is sufficiently exact:

$$p\,v = .62\,M\,T, \qquad \textbf{(119.)}$$

where M is the weight of gas and T is the absolute temperature.

Corresponding to formulas **81, 86,** and **87,** *Heat*, we have the following approximate formulas for the *adiabatic* expansion of superheated ammonia:

$$p_1 V_1^{1.3} = p_2 V_2^{1.3}. \qquad \textbf{(120.)}$$

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{.3}{1.3}} = \left(\frac{p_1}{p_2}\right)^{.2308}. \qquad \textbf{(121.)}$$

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{.3}. \qquad \textbf{(122.)}$$

1361. Aqua Ammonia.—As already stated (Art. **1356**), aqua ammonia, known also as **ammonia liquor,** is a solution of ammonia gas in water. At 32° F. and under atmospheric pressure, water will absorb 1,140 times its volume of ammonia gas. The amount of gas held in solution affects the specific gravity of the solution; the more gas absorbed the less the density. The amount of ammonia that can be absorbed by water is governed by the temperature of the water and the pressure of the gas. The colder the water and greater the pressure, the greater the quantity of ammonia taken up.

1362. The strength of a solution of anhydrous ammonia in water is determined by an instrument called a **hydrometer**—Beaumé's hydrometer—which is used for determining the densities of various liquids, as shown in Fig. 321. It consists of a glass tube with a bulb near the middle and a second bulb near the end, partly filled with mercury. A graduated scale is marked on the stem. When this instrument is placed in a liquid, it is evident that it will sink deeper the less the density of the liquid; hence the density will be indicated by the mark on the scale at the level of the liquid. For liquids lighter than water, the point to which the instrument sinks when placed in a solution of 10 parts of salt to 90 of water is marked 0°, and the point to which it sinks in distilled water is marked 10°. The space between the two marks is divided into 10 parts, and the division is continued to the top of the stem. The hydrometer thus graduated is generally used for ammonia solutions, though there is another graduation in which the reading for pure water is 0° instead of 10°.

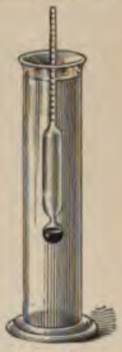


FIG. 321.

In Table 27, the first column gives the number of parts of ammonia gas in 100 parts of the solution, the second column gives the specific gravity of the solution, and the third column gives the corresponding reading on the Beaumé hydrometer. For example, if the hydrometer reading is 16°, the solution consists of 10 parts, by weight, of ammonia to 90 parts of water, and the specific gravity of the solution is .960.

1363. Heat of Absorption.—As stated in *Heat*, all chemical actions are accompanied by an increase or decrease

TABLE 27.

STRENGTH OF AMMONIA LIQUOR.

Percentage of Ammonia by Weight.	Specific Gravity.	Degrees Beaumé.	
		Water 10°.	Water 0°.
0	1.000	10.0	0
1	.993	11.0	1.0
2	.986	12.0	2.0
4	.979	13.0	3.0
6	.972	14.0	4.0
8	.966	15.0	5.0
10	.960	16.0	6.0
12	.953	17.1	7.0
14	.945	18.3	8.2
16	.938	19.5	9.2
18	.931	20.7	10.3
20	.925	21.7	11.2
22	.919	22.8	12.3
24	.913	23.9	13.2
26	.907	24.8	14.3
28	.902	25.7	15.2
30	.897	26.6	16.2
32	.892	27.5	17.3
34	.888	28.4	18.2
36	.884	29.3	19.1
38	.880	30.2	20.0

in the temperature of the mixture. This is especially true of solutions. In the case of ammonia absorbed in water, 925.7 B. T. U. is given up for each pound of ammonia gas absorbed under atmospheric pressure. Though no very exhaustive experiments have been made on this subject, results deduced from the practical running of refrigerating-machines show that this figure is practically constant.

Since heat is given up when ammonia gas is absorbed, heat will be absorbed when the gas is again liberated from the water. The quantity of heat necessary to liberate one pound of anhydrous gas is 925.7 B. T. U., the same amount of heat that is given out by the solution when the gas is being absorbed.

1364. Tests for Ammonia.—If it is desired to test the purity of liquid anhydrous ammonia, draw some out into a flask having a cork with a bent tube inserted in it. Wrap the flask up in dry waste or cloth before drawing off the ammonia, or the fingers are liable to be frozen fast to the flask. The liquid ammonia evaporates slowly, the gas passing out of the bent tube. If an accurate low-temperature thermometer is obtainable and is immersed in the boiling liquid, it should indicate a temperature of −28.5° F., with normal barometric pressure. If the liquid is pure anhydrous ammonia, there should be no residue left in the flask. A deposit of oil or water indicates impure ammonia.

To detect a leak in piping in case the odor does not betray it, hold a glass rod moistened with muriatic acid near the supposed leak. A white fume rising from the rod indicates an escape of ammonia.

To detect ammonia leaks in piping under water or brine, add to a sample of the suspected liquid a few drops of *Nessler's Reagent*; a yellow coloring indicates traces of ammonia, but if the quantity of ammonia is large, the color changes to a dark brown.

1365. To Prepare "Nessler's Reagent."—Dissolve 17 grams (0.6 oz.) of mercuric chloride in about 300 cubic centimeters (10½ fluid oz.) of distilled water; dissolve 35 grams (1¼ oz.) of potassium iodide in 100 cubic centimeters (3½ oz.) of water; add the former solution to the latter, with constant stirring, until a slight permanent red precipitate is produced. Next dissolve 120 grams (4¼ oz.) of potassium hydrate in about 200 cubic centimeters (7 oz.) of water; allow the solution to cool; add it to the above solution, and make up with water to one liter (33¾ oz.), then add mercuric chloride solution until a permanent precipitate again forms; allow to stand till settled, and decant off the clear solution for use; keep it in glass-stoppered blue bottles, and set away in a dark place to keep it from decomposing.

RELATIVE EFFECT OF REFRIGERATING FLUIDS.

1366. The density of the gas at the evaporating temperature and the latent heat of the liquid determine the size of the compression cylinder necessary for any required capacity. The same machine working between 5° and 64.4° will give the following cooling effects per *cubic foot* of compressor piston displacement under theoretically perfect conditions:

Carbon Dioxide	248.18 B. T. U.
Ammonia	62.75 B. T. U.
Sulphur Dioxide	22.88 B. T. U.
Sulphuric Ether	3.68 B. T. U.

THE AMMONIA COMPRESSION SYSTEM.

GENERAL DESCRIPTION.

1367. Suppose a flask or ordinary bottle *B*, Fig. 322, supplied with a cork having a bent tube *G* inserted, be partially filled with anhydrous ammonia. This can be done easily, as the evaporation of the ammonia is comparatively slow, owing to its high latent heat. As the ammonia enters the flask, frost will begin to gather on the outside. If, now, we place this flask into a pail *A* partially filled with water *C*, in a short time ice *D* will begin to gather on the outside of the flask. This is the simplest form of ice-machine, but in this form the liquid ammonia, when it evaporates, passes out of the flask and is lost.

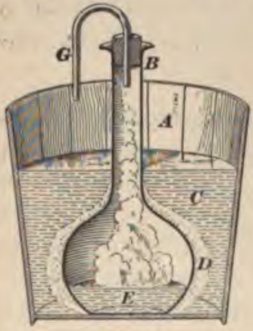


FIG. 322.

As with all volatile vapors, the temperature at which vaporization (or condensation) occurs rises as the pressure of the vapor increases. To prove this, insert a thermometer into the flask so that the bulb is immersed in the boiling ammonia. The temperature will fall rapidly, and if the thermometer is correct, it should register 28.5° below zero. Take a piece of pipe; weld or plug one end and fit the other with a cap *B*, Fig. 323. Arrange a stuffing-box *C* about the thermometer *T* in the cap; also provide an opening connecting with the pressure gauge *P*. Unscrew the cap and pour the contents of the flask into the pipe *A* and screw on the cap *B*. If the gauge points to zero, the thermometer should still read −28.5° F. Watch the gauge and thermometer carefully. The ammonia evaporating in the pipe liberates gas. As this gas can not escape, it creates a pressure in the pipe, which will be shown on the gauge, and a corresponding increase in the temperature of the boiling ammonia will become apparent. This will continue until the temperature of the liquid will be identical with the surrounding objects. Assume this temperature to be about 70° F.; the gauge should then show a pressure of 130 pounds per square inch. If, therefore, we keep the temperature of the pipe at 70° by immersing it in water at that temperature, and arrange to keep a pressure slightly in excess of 130 pounds per square inch in the pipe, no further evaporation will take place, and the remaining liquid ammonia will lie quietly in the pipe.

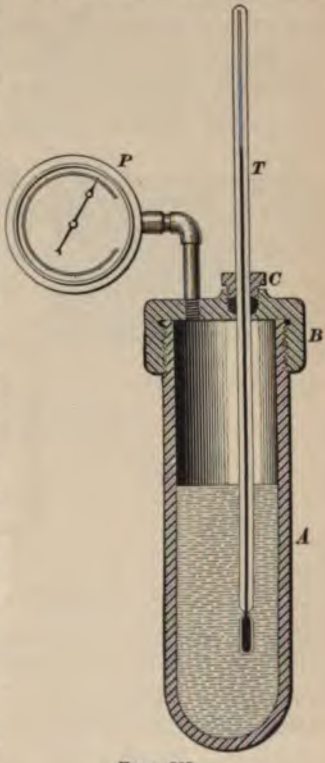


FIG. 323.

1368. From the foregoing, it is apparent that if some means be devised of taking the evaporating gas as it leaves the flask in Fig. 322 and transfer this gas into the pipe of Fig. 323, it would be possible to save the gas. In place of a short piece of pipe *A*, Fig. 323, take a large coil of pipe *A*, Fig. 324, submerged in a tank of water *C*. The water enters by means of the pipe *F*, and the overflow passes out of the pipe *F'*; the continuous flow tends to keep the temperature of the coil *A* constant. Replace the flask *B* in Fig. 322 with a coil of pipe *B*, Fig. 324, immersed in a water-tank *D*. Provide a pump capable of working against a high pressure, and connect the suction of the pump with the coil *B* and the discharge with the coil *A*; also provide pressure gauges *G'* and *G* on each of these lines. Connect the bottom of the two coils together, and place a valve *E* in the line. Partially fill the coil *A* with anhydrous ammonia. If the temperature of the water in *A* is about 70°, the gauge *G'* will show a pressure of 130 pounds. Open the valve *E* slightly and leave it open. The pressure denoted by the gauge *G* will gradually rise, and ice will begin to form on the lower pipes of *B*. When the pressure shown by *G* has reached 15 pounds, start the pump *P*. This pump will draw the gas out of the coil *B*, compress it, and deliver it to the coil *A*. The gas entering *A*, which has been heated by the

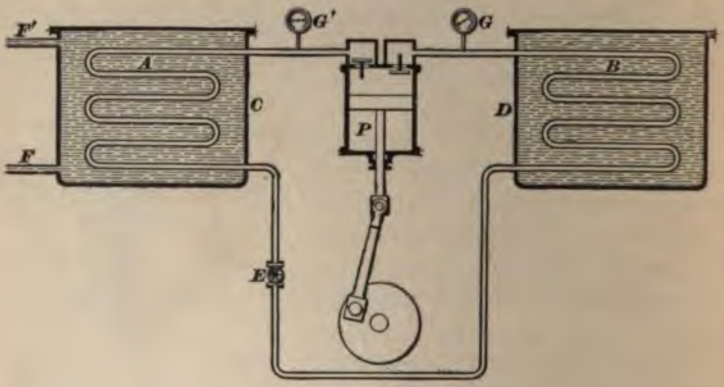


FIG. 324.

compression, comes in contact with the cold pipe surface, and is first cooled until its temperature is but little above that of the condensing water flowing out through F'. The gas then condenses and falls to the bottom of the coil in the form of liquid anhydrous ammonia. As the valve E is open, the coil A is prevented from filling up. The withdrawal of a quantity of the gas in the coil B tends to decrease the pressure in that coil; however, a quantity of the liquid passes from A to B through the expansion-valve E, vaporizes, and supplies an amount of gas equal to that withdrawn by the pump.

THE CYCLE OF THE AMMONIA COMPRESSION MACHINE.

1369. In Fig. 325 let the length $O V_1$ represent to scale the volume of the compressor cylinder P, Fig. 324,

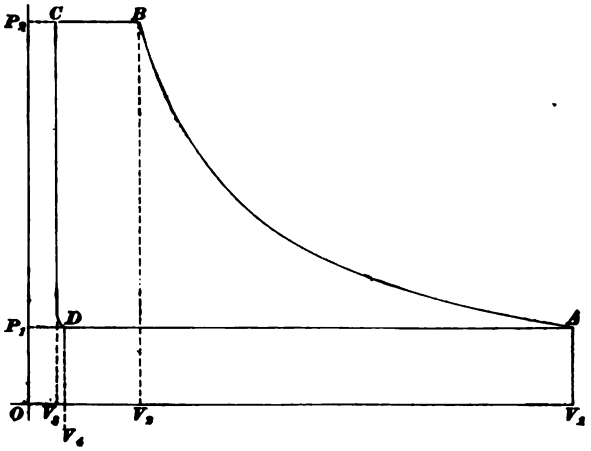


FIG. 325.

and let the ordinate $O P_1$ $(= V_1 A)$ represent the pressure of the ammonia gas in the coil B. Then we may say that the point A represents the state of the gas in the cylinder when the piston is at the lower end of the cylinder. As the piston rises, the gas is compressed until it reaches the state represented by the point B; that is, the pressure is represented by $O P_2$ and the volume by $O V_2$. The character of the compression depends upon the method of cooling the compression cylinder, to be explained later. We will assume for the present that the compression is adiabatic. If the vapor is dry and saturated at the state A, it will be superheated at the state B, after the adiabatic compression. When the pressure in the cylinder reaches P_2, the valve opens and the compressed vapor is delivered to the coil A at a practically constant pressure. During this operation the temperature of the superheated vapor is reduced, and then the vapor is wholly or partially condensed to a liquid. The volume is reduced from $O\,V_2$ to $O\,V_3$. The ammonia liquid now runs slowly through the valve E into the coil B. Since the pressure in the coil B is much lower than that in A, there will be a drop in pressure in passing the valve, and this will be accompanied by a drop in temperature. The slight increase in volume from V_3 to V_4 is due to the fact that a small quantity of the liquid is vaporized before the pressure has dropped to P_1. The liquid ammonia in the state represented by the point D now vaporizes, and the volume increases from $O\,V_4$ to $O\,V_1$. The gas returns to its original state represented by A, and the cycle is closed.

The cycle just described differs from the ideal cycle described in Art. **1335** in having the line $C\,D$, representing the third operation, nearly straight instead of curved, as in Fig. 317. To attain the ideal conditions shown in Fig. 317, it would be necessary to allow the liquid from the condenser A to expand and do work in an expansion cylinder instead of passing through the regulating valve E. For constructive reasons, the valve is preferred, as the cylinder would be small, and the work recovered by the expansion would in any case be insignificant.

GENERAL THEORY.

1370. Heat Transfers.—Let T_a denote the absolute temperature of the gas in the state A, Fig. 325, which is practically the temperature of the coil B, and let T_b

denote the temperature of the gas at the end of compression. Since the gas is superheated during the compression, the final temperature T_b is given by formula **121:**

$$\frac{T_b}{T_a} = \left(\frac{p_2}{p_1}\right)^{.2308},$$

or

$$T_b = T_a\left(\frac{p_2}{p_1}\right)^{.2308}.$$

Let T_c denote the final temperature that the gas or vapor attains in the condenser; that is, T_c is the temperature of condensation. Let r_2 denote the latent heat of vaporization at the temperature T_c. The specific heat of ammonia gas being .508, the heat given up by a pound of the gas in cooling from T_b to T_c is .508 $(T_b - T_c)$ B. T. U. In condensing, a pound of the gas gives up further r_2 B. T. U. If M denote the weight in pounds of gas used per stroke, then the heat given up to the condenser per stroke is evidently

$$\left.\begin{aligned} Q_1 &= M[.508\,(T_b - T_c) + r_2]\text{ B. T. U.} \\ &= M[.508\,(t_b - t_c) + r_2]\text{ B. T. U.} \end{aligned}\right\} \quad (a)$$

If t_e and t_f denote, respectively, the temperature of the cooling water as it enters and as it leaves the condenser, then the weight of cooling water required per stroke is

$$G = \frac{M[.508\,(t_b - t_c) + r_2]}{t_f - t_e}. \quad (b)$$

For the sake of simplicity, we will assume that the pressure of the liquid ammonia drops from P_2 to P_1 before vaporization begins; while this is not precisely the case, the difference in the result is not appreciable. During the fall of temperature from T_c to T_a, the original temperature of the gas in the coil B, a pound of the liquid gives up $s\,(T_c - T_a)$ heat-units, where s denotes the specific heat of liquid ammonia. During the vaporization in the coil B, each pound absorbs r_1 B. T. U., where r_1 denotes the latent heat of vaporization corresponding to the pressure P_1 and temperature T_a. The net heat abstracted from the refrigerator per stroke is therefore

$$\left.\begin{aligned} Q_2 &= M[r_1 - s(T_c - T_a)] \text{ B. T. U.} \\ &= M[r_1 - s(t_c - t_a)] \text{ B. T. U.} \end{aligned}\right\} \quad (c)$$

Since the specific heat of liquid ammonia is practically 1, we may write the equation:

$$\left.\begin{aligned} Q_2 &= M[r_1 - (T_c - T_a)] \text{ B. T. U.} \\ &= M[r_1 - (t_c - t_a)] \text{ B. T. U.} \end{aligned}\right\} \quad (d)$$

The work of the compressor per stroke in foot-pounds is

$$W = J(Q_1 - Q_2) = J\,M\,[r_2 - r_1 + .508\,(t_b - t_c) + t_c - t_a].$$

If n denote the number of strokes per minute, the theoretical horsepower of the compressor is

$$H = \frac{n\,J\,M[r_2 - r_1 + .508\,(t_b - t_c) + t_c - t_a]}{33{,}000}. \quad (e)$$

1371. Efficiency.—The theoretical efficiency of the refrigerating-machine is

$$E = \frac{J\,Q_2}{W} = \frac{r_1 - (T_c - T_a)}{r_2 - r_1 + .508\,(T_b - T_c) + T_c - T_a}. \quad (f)$$

If we denote by T_i the temperature of the cold room or of the brine, if the latter be used, then the effective range of temperature is $T_f - T_i$, where T_f, as before, denotes the absolute temperature of the cooling water as it leaves the condenser. In practice T_i is always a little higher (5° to 10°) than the temperature T_a of the ammonia in the refrigerating coils, and the temperature T_c of the ammonia in the condenser coils is always higher than T_f. The theoretical maximum efficiency for this temperature range is

$$E_m = \frac{T_i}{T_f - T_i}. \quad (g) \qquad \text{(See Art. } \mathbf{1346.}\text{)}$$

The economy of a machine may be judged by comparing the actual efficiency with this ideal efficiency E_m.

1372. Volume of Compressor Cylinder.—Let v denote the volume of a pound of ammonia vapor at the pressure p_1 in the coil B; then, since M pounds of vapor are used per stroke, the cubic capacity of the cylinder must be

$$C = M v \text{ cubic feet.} \quad (h)$$

To allow for clearance and imperfections in the operation, the volume as thus calculated should be increased 10 per cent. or more.

1373. Capacity.—In 24 hours the heat abstracted from the cold body is

$$(24 \times 60 \times n \times Q_2) \text{ B. T. U.} =$$
$$24 \times 60 \times n M [r_1 - (t_c - t_a)] \text{ B. T. U.}$$

Hence, the ice-melting capacity in tons per 24 hours is

$$F = \frac{24 \times 60 \times n M [r_1 - (t_c - t_a)]}{285{,}300} =$$
$$.00505 n M [r_1 - (t_c - t_a)]. \qquad \textbf{(123.)}$$

1374. Problem.—As an application of the theory developed in the preceding paragraphs, the dimensions of an ammonia refrigerating-machine will be worked out from the following data:

Ice-melting capacity.............. 20 tons in 24 hours;
Temperature in condenser......... 90° F.;
Temperature in refrigerating coil.. 15° F.;
Initial and final temperatures of condensing water, 60° and 85° F.

The compressor is double-acting and makes 56 revolutions per minute. Required, also, the horsepower of the compressor, the theoretical efficiency, and the quantity of cooling water used per minute.

Referring to the table of the properties of ammonia, the pressures corresponding to the temperatures 15° and 90° F. are, respectively, $p_1 = 42.93$ pounds and $p_2 = 182.8$ pounds, absolute, per square inch. The latent heats are, respectively, $r_1 = 546.26$ B. T. U. and $r_2 = 498.11$ B. T. U. The absolute temperature T_a is 460° + 15° = 475°; the absolute temperature T_b of the superheated gas at the end of compression is

$$T_b = T_a \left(\frac{p_2}{p_1}\right)^{.2308} = 475° \times \left(\frac{182.8}{42.93}\right)^{.2308} = 663.62°;$$

hence, $$t_b = 663.62° - 460° = 203.62°.$$

The capacity being 20 tons, we have

$$20 = .00505\, n\, M\, [r_1 - (t_e - t_a)\,],$$

or $$M = \frac{20}{.00505\, n\, [r_1 - (t_e - t_a)]} =$$

$$\frac{20}{.00505 \times 112 \times [546.26 - (90 - 15)]} = .075.$$

The quantity of ammonia used per stroke is .075 pound, and the quantity circulated per minute is .075 lb. × 112 = 8.4 pounds.

The volume of 1 pound of ammonia vapor at a temperature of 15° is, according to the table, 6.49 cubic feet; hence, the theoretical capacity of the cylinder is

$$C = Mv = .075 \times 6.49 \text{ cu. ft.} = .48675 \text{ cu. ft.} = 841.2 \text{ cu. in.}$$

Adding $\frac{1}{8}$ to allow for imperfections in the operation, the actual volume is 841.2 cu. in. × $1\frac{1}{8}$ = 946.5 cu. in. If the stroke is made double the piston diameter, this volume will require a diameter of $8\frac{1}{2}$ inches and a stroke of 17 inches.

The horsepower required to drive the compressor is, approximately,

$$H = \frac{nJM\,[r_2 - r_1 + .508\,(t_b - t_c) + t_c - t_a]}{33{,}000} =$$

$$\frac{112 \times 778 \times .075 \times [498.11 - 546.26 + .508\,(203.62 - 90) + 90 - 15]}{33{,}000} =$$

16.7, nearly.

The horsepower of the steam cylinder should be at least $\frac{1}{3}$ greater than that of the compressor; in the present case it will be about 16.7 H. P. × $\frac{4}{3}$ = 22.93, say 23 H. P.

The heat given up by the ammonia to the cooling water per stroke is

$$Q_1 = M\,[.508\,(t_b - t_c) + r_2]\text{ B. T. U.} =$$
$$.075\,[.508\,(203.62 - 90) + 498.11]\text{ B. T. U.} = 41.687\text{ B. T. U.}$$

The cooling water required per minute is therefore

$$G = \frac{n\,Q_1}{t_f - t_e} = \frac{112 \times 41.687}{85 - 60} = 187\text{ lb., nearly.}$$

187 pounds is equivalent to 3 cubic feet, or about $22\frac{1}{2}$ gallons.

The theoretical efficiency is

$$E = \frac{r_1 - (t_c - t_a)}{r_2 - r_1 + .508\,(t_b - t_c) + t_c - t_a} = \frac{471.26}{84.569} = 5.57.$$

If we assume the temperature of the brine or of the cold room to be 5° higher than that of the ammonia in the condenser, $T_i = T_a + 5° = 460° + 15° + 5° = 480°$. The effective temperature range is therefore $T_f - T_i = 85° + 460° - 480° = 65°$, and the maximum efficiency is

$$E_m = \frac{T_i}{T_f - T_i} = \frac{480}{65} = 7.38.$$

1375. Influence of Suction Pressure.—The *suction* pressure, that is, the pressure of the ammonia vapor as it passes from the refrigerating coil to the compressor cylinder, exerts a marked influence upon the capacity and also upon the economy of the refrigerating-machine. Consider the formula for capacity, $F = .00505\,n\,M\,[r_1 - (t_c - t_a)]$. Since the n remains constant, it is evident that the capacity depends upon the factors M and $[r_1 - (t_c - t_a)]$. Referring to the table of properties of ammonia, it is seen that as the suction pressure p_1 is increased, the corresponding latent heat r_1 becomes less and the corresponding temperature t_a rises; consequently the factor $[r_1 - (t_c - t_a)]$ changes but little with different suction pressures, and *the capacity is practically proportional to the weight of ammonia circulated*, that is, to M. Now, since the same *volume* of ammonia vapor passes through the compressor in a given time, whatever the pressure may be, it is evident that the *weight* of ammonia circulated is exactly proportional to the weight of the vapor per cubic foot. Thus, referring to the table, a cubic foot of vapor at a pressure of 20.99 pounds per square inch absolute weighs .0779 pound, while at a pressure of 42.93 pounds absolute a cubic foot weighs .154 pound, or practically twice as much. With the greater suction pressure, therefore, the machine will circulate double the ammonia that it will with the lower suction pressure, and consequently the capacity at the higher pressure will be nearly double that at the lower pressure.

The power required to operate the compressor is of course greater with the higher suction pressure and greater capacity, but the increase in power is not proportional to the increase in capacity. It can readily be shown that the ratio of the power consumed to the capacity decreases as the suction pressure is raised. The power is proportional to $Q_1 - Q_2$ (see Art. **1370**), and the capacity is proportional to Q_2 (see Art. **1373**); hence the ratio $\frac{H}{F}$ is proportional to $\frac{Q_1 - Q_2}{Q_2}$. Now, taking the expressions for Q_1 and Q_2, $Q_1 = M[.508(t_b - t_c) + r_2]$ and $Q_2 = M[r_1 - (t_c - t_a)]$, it is seen that the only effect on Q_1 of raising the suction pressure is to lower the temperature t_b at the end of compression. The effect on Q_2 is to decrease r_1 and increase t_a; but since t_a increases faster than r_1 decreases, the net effect of raising the suction pressure is to increase Q_2. Since Q_1 is decreased and Q_2 is increased, the fraction $\frac{Q_1 - Q_2}{Q_2}$, and consequently the ratio $\frac{H}{F}$, is decreased by raising the suction pressure. We have, therefore, the important fact that *increasing the suction pressure increases* ***both*** *the* ***capacity*** *and* ***economy*** *of the machine.*

Fig. 326 is the indicator-diagram taken from an ammonia cylinder of a compressor running under 151 pounds head pressure and 28 pounds suction pressure. The tonnage in this case is 74.8, while the ammonia cylinder horsepower is but 65.7 or .88 of a horsepower per ton of work done. Fig. 327 shows a diagram taken from the same compressor under the following conditions: Head pressure 135 pounds, back pressure 2 pounds, ammonia cylinder horsepower 46.02; tonnage 25.9 or 1.8 horsepower per ton of refrigerating effect. From this it will be seen that while the capacity is about a third, the expenditure of power per ton of refrigerating effect is approximately double. These diagrams and the data connected with them are taken from Prof. J. E. Denton's report of a test of a refrigerating plant. (Transactions A. S. M. E., Vol. XII.)

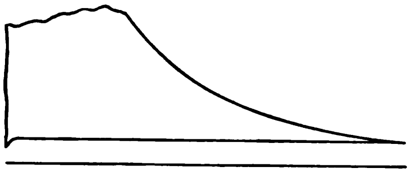


FIG. 326.

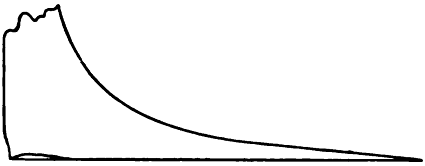


FIG. 327.

The upper limit of suction pressure is fixed by the temperature at which it is required to keep the cold room or the circulating brine. Suppose, for example, that the average temperature of the brine is 33° F.; the temperature in the refrigerating coils must be, say, 10° lower, or 23° F., and the pressure corresponding is 51.25 pounds per square inch absolute, or 36.55 pounds gauge.

1376. Influence of Condensing Pressure.—The pressure in the condenser does not affect the weight of ammonia circulated, and has therefore no influence on the capacity of the refrigerating-machine. The lower the temperature of the ammonia in the condenser can be kept, the lower will be the pressure, and consequently the less will be the work of compression. It is advantageous, therefore, to have the initial temperature of the condensing water as low as possible and to use as much water as is consistent with economy. Prof. Denton, in the report previously mentioned, states that "maximum economy with a given type of engine, where water must be bought at average city prices, is obtained at 28 pounds suction pressure and 150 pounds condensing pressure." This condensing pressure requires about a gallon of water per minute per ton of ice-melting capacity, taking the initial temperature of the water at 56° F. If the supply of water is 25 per cent. less, the condensing pressure rises to 190 pounds and the work of compression is increased about 20 per cent. On the other hand, if the water-supply is increased to three gallons per minute per ton capacity, the condensing pressure will fall to about 105 pounds. The work of compression is reduced about 25 per cent., but the saving in power thus effected is more than counterbalanced by the cost of the extra water used to lower the pressure.

THE COMPRESSOR OR PUMP.

1377. The vital part of a compression refrigerating-machine is the compressor or pump; upon the design and construction of this part chiefly rests the commercial advantage or disadvantage of any particular make of machine.

1378. Pump Cylinder.—Compressor cylinders are usually made of a fine, close-grained charcoal iron or semi-steel. It is necessary to have a cylinder heavy enough to withstand a test pressure of at least 350 pounds per square inch. Heads and bolts should be made in proportion. The joint between the cylinder and head is usually of a tongue and groove, male and female type, lead being used as a packing material in the groove. Other makers use a smooth face on the flanges of the head and cylinder and a very thin lead or other metallic packing. In this case, the flange should be scored with three or four concentric grooves between the bore of cylinder and bolt-holes, to hold the gasket in place and prevent it from blowing out. The bore or inside walls of the cylinder should be smooth, and sufficient oil must be used in running the machine to prevent groaning and consequent wear.

Some makers line the cylinder with a wrought-iron or steel bushing, but the majority use cast iron. There is more friction with the former metals, while with care and a proper mixture, a cast-iron cylinder that will hold ammonia can be made.

The stroke should in all cases be as long as possible, so as to get a comparative fast piston speed with the least number of revolutions. Another reason for making the stroke long is that the clearance spaces are smaller with a long stroke and small bore than a shorter stroke and longer bore. A well-proportioned compressor should have a stroke of at least double the diameter of the bore, and two and one-half times is better practice. The following cylinder capacities will be found convenient for ready reference: With a suction or back pressure of 25 pounds, a single-acting compressor should displace 3½ cubic feet per ton and a double-acting compressor 4 cubic feet per ton. With 15 pounds back pressure, these figures should be 5 cubic feet for the former and 5½ cubic feet for the latter type of machine. These figures are for well-built compressors having comparatively small clearances, and are based upon actual tests of refrigerating-machines.

1379. Compressor Piston.—The pistons of all double-acting compressors are necessarily solid. Those of the single-acting type are often made solid, while others are fitted with suction-valves. Cast-iron spring rings are used for packing-rings to prevent leakage; at least 4 or 5 rings should be employed. Sectional rings and others similar to those used in high-class engines and air-compressors are found in ammonia pumps.

1380. Heat of Compression.—It was stated in Art. **1369** that if ammonia vapor be compressed adiabatically, it will be superheated, and the work done on the vapor by the piston will be stored up in the vapor in the form of heat. This heat must be gotten rid of during the period of compression, otherwise it must be absorbed by the condensing water before the vapor condenses. It is quite evident that it will be most economical to remove the heat, as far as possible, as fast as it is generated, and keep the temperature of the cylinder comparatively low throughout the compression. In fact, it is absolutely necessary to employ some method of keeping the cylinder cool, otherwise the excessive heat developed in compression would soon become so great that the gas would enter the cylinder in a greatly superheated state, which would lessen its density. This decrease in density would naturally cause a corresponding decrease in the weight of gas pumped in a given time, thus affecting both capacity and economy.

A number of expedients have been tried with this object in view. The simplest of these is jacketing the cylinder with water. This method of cooling the compression cylinder is known as the **dry-compression system.** The gas enters the cylinder in perhaps a saturated state, though usually somewhat superheated; the instant compression begins, however, the vapor is immediately superheated.

In the second method, **wet compression,** the cylinder is not jacketed, but a certain amount of liquid anhydrous ammonia is allowed to enter the cylinder with each stroke of the compressor; the mixture of vapor and liquid remains saturated while it is compressed, the heat equivalent of the work of compression is taken up by the vaporization of a part of the liquid, and the vapor remains at the temperature due to the pressure.

The third method employed is a modification of wet compression. Instead of permitting anhydrous ammonia to enter the cylinder, a certain quantity of oil is injected during the stroke; the purpose of the oil is to cool the gas during compression and seal the valves so as to cut down the clearance space.

1381. Dry Compression.—The majority of compressors built in the United States are of the water-jacketed, dry-compression type. In the case of vertical compressors, the water-jackets are merely small tanks enclosing the walls of the cylinder, and are sufficiently high so that the top head of the cylinder is also immersed. They are open at the top and the water passes off by gravity. Horizontal compressors are usually water-jacketed on the cylinder walls only, the heads being unjacketed. With the inlet water at 60° F., the quantity of water required for a water-jacket is about $\frac{1}{5}$ of a gallon per minute per ton of refrigerating effect.

1382. Wet Compression.—The injection of a small quantity of anhydrous ammonia to cool by its evaporation the walls of the cylinder was the invention of Prof. C. P. G. Linde, of Munich, Germany. The machines built under this system bear his name and are of the horizontal, double-acting type. The temperature of the gas leaving the compressor in case of the Linde machine is much lower than that in the dry-compression system, and the theoretical economy is somewhat higher than that of the latter system. An objection sometimes urged against wet compression is the necessity of introducing a small quantity of liquid ammonia, which if increased in any degree by carelessness is liable to act somewhat like water carried over in the steam of a steam engine, and may blow out the cylinder-head. The chances of such an occurrence are so small that the objection does not appear to be a serious one.

1383. Oil Injection.—The third method of cooling the cylinder during compression is that of the De La Vergne Company. In the earlier machines a small quantity of oil was admitted into the compression cylinder during suction and was expelled at compression. The mixture of oil and gas at a somewhat high temperature passed to the oil separator, where the oil was separated from the ammonia gas. The gas passed on to the condenser in its regular cycle; the oil was taken from the separator, passed through a cooling coil immersed in running water, and was then allowed to run into a receiver, from which it again passed into the suction-pipe.

It will be seen from this description that as a certain quantity of oil was allowed to enter with the gas, the volume of the gas entering the cylinder at each stroke was decreased in proportion to the amount of oil injected; in case a considerable quantity of oil was fed in, this would cut down the capacity appreciably. In order to obviate this difficulty, the De La Vergne Company in their new compressors inject oil by means of a small pump after the work of compression has set in, and not during suction as formerly. This also permits the oil to be kept fully charged with ammonia.

1384. Single and Double Acting Compressors.—The single-acting ammonia compressor is usually of the vertical type and compresses gas on one side of the piston only, viz., the upper side, or, in other words, the side farthest away from the stuffing-box. The gas has at all times access to the space in the cylinder below the piston; here only the suction pressure acts constantly on the under side of the piston. In this form of compressor, the pressure on the stuffing-box side is always low, and it is therefore easy to keep the piston rod packed. As the work of compression is done on one side of the piston only, the water-jacket has a greater opportunity of reducing the temperature of the cylinder walls. The colder the compressor cylinder is kept, the greater will be the weight of gas pumped per revolution, as explained in previous paragraphs. It will therefore be readily seen that the loss due to superheating is less in this form of compressor than in the double-acting type. Experiment and test have proven this to be the case, the loss due to superheating in single-acting compressors being from 15 to 22 per cent.

The double-acting compressor is usually of the horizontal type and has suction and delivery valves on both sides of the piston. Such a compressor is cheaper to build than the single-acting one, and with the same cylinder capacity will deliver double the quantity of ammonia. The loss due to cylinder superheating, however, is greater than in the single-acting type, this loss amounting to 25 or 30 per cent.

The clearance spaces in a single-acting are usually much smaller than those of the double-acting compressor. In the former type it is merely necessary to adjust the piston so that it just clears the upper head of the compressor; this can be done, no matter what the stroke of the pump may be. In the case of the double-acting machine, however, the adjustment is a much more difficult task, for if the clearance is cut down on one side of the piston, it will increase by a like amount on the other side. It is therefore necessary to be very careful, in the selection of gaskets, etc., to get them of the proper thickness so that the clearance spaces will be reduced to the minimum on both sides of the piston.

1385. Vertical and Horizontal Compressors.—In all types of compressors except those using oil injection, as little oil is fed into the compression cylinder as possible. This small quantity of oil, which is much less than that used in the case of steam engines, makes it necessary to select a compressor in which the wear on the moving parts is equal all around. In case of a vertical compressor, the weight of the piston does not come on the walls of the cylinder, as is the case with the horizontal machine. The valves are also usually arranged so that they work up and down, and will therefore not wear their stems in such a way as to get them out of line with their seats. The vertical machine, however, is more expensive to build than the ordinary horizontal compressor, the bed-plate of which has the form of that of an ordinary horizontal engine. A still cheaper construction is that of the direct-acting horizontal machine, in which the compressor is attached tandem with the engine cylinder on an extension of the steam-engine bed-plate. The advantages of horizontal compressors are that they can be placed in low headroom and that all parts are accessible from the floor. A *direct-acting* machine is one in which the compressor and steam cylinders are in line, and the compressor piston and steam piston are attached to the same piston rod. In the *indirect* machine, the steam and compressor cylinders have separate piston rods and connecting-rods; the compressor is driven from a main shaft, which is in turn driven by the steam engine in the ordinary manner.

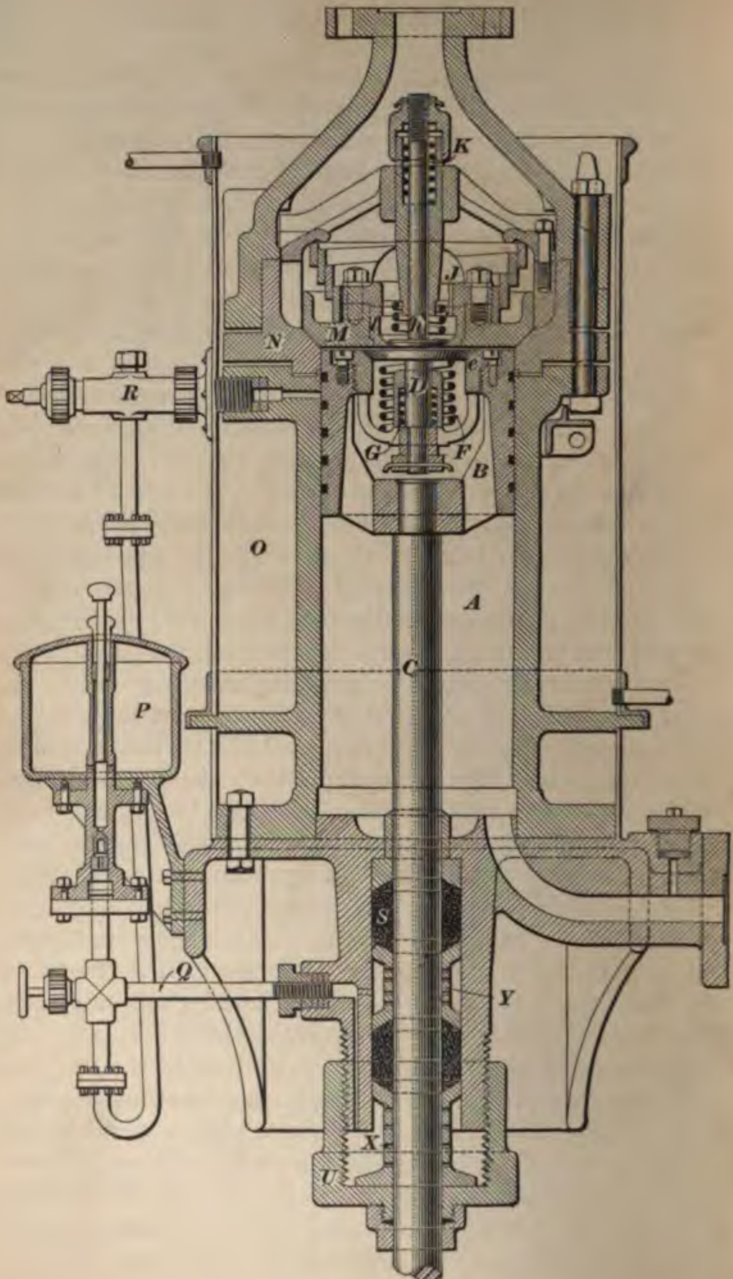


FIG. 328.

1386. Clearance Spaces.—When a pump piston is at the end of its stroke, the space that is left between the piston, cylinder-head, and valves is known as the clearance space. If this space is reduced to nothing, that is, if the valves are so ground that they come flush with the head, and the piston is allowed to come up until it just touches the head, all gas between the piston and head will be expelled through the delivery valve. Such a construction is hardly practical. It will be seen, however, that it is necessary to make this space as small as possible, since the gas that is left in the cylinder will expand as the piston travels back and will keep a corresponding quantity of gas from entering the cylinder.

Well-made compressors are run with a clearance of $\frac{1}{32}$ inch between the piston and cylinder-head. In order to determine this clearance, one of the valves should be taken out and a piece of fuse wire or lead laid on top of the piston. The machine is then turned by hand until the piston has completed its stroke. The fuse wire or lead will have been flattened, and its thickness will be the exact distance between the cylinder-head and piston, and will thus indicate the amount of clearance. The piston rod can be adjusted if the first test shows an unsatisfactory clearance, and the experiment may be repeated until the clearance space is cut down to a minimum.

1387. Suction and Discharge Valves.—A section through a compressor cylinder as constructed by the Frick Company is shown in Fig. 328. The suction-valve is shown at *D*. As the piston moves downwards, the valve rises from its seat *e*, and the gas from the refrigerating coil, which has at all times free access to the lower end of the cylinder *A*, passes through the open valve to the upper end of the cylinder. As soon as the piston again starts upwards, the valve *D* closes and compression begins. The valve is fitted with balanced springs *F* and *G*, which may be adjusted so as to prevent the valve from rattling.

Ice-machine builders have for a long time been striving to design a suction-valve so arranged that in case either the valve-stem or valve breaks, the pieces will not fall into the cylinder of the compressor. Compressors built in former years were greatly troubled with imperfect workmanship on their suction-valves. Machines were occasionally wrecked because of a broken valve. The fragments dropped into the cylinder, and the piston on the next stroke drove them against the head, thus causing serious damage.

The discharge-valve is an ordinary valve with a spindle *k* and with balanced springs *J* and *K*. In the construction shown in Fig. 328, the whole head *M* is a delivery-valve. Under ordinary circumstances, the gas passes through the small discharge-valve, but in case the suction-valve is displaced by any accident, the head *M* will rise from its seat. This action prevents damage to the head.

The suction and delivery valves and also the valve-seats are made of steel. The valves should be well ground to their seats, and they should be inspected occasionally to ascertain if either valve or seat is cut or defective in any way. A spare set of valves should always be carried in stock to replace any that may be found defective.

The area of the suction-valve should be at least equal to the area of the suction-pipe. The lift of the valve should be at least ¼ of the diameter, in order that the area of the annular opening may be equal to the area of the valve.

1388. The Stuffing-Box.—As shown in Fig. 328, the stuffing-box *S* is made very long in order that a large amount of packing may be used. The stuffing-box is usually provided with a thimble or oil-gland *Y* placed about midway between the two ends. A certain amount of packing is first inserted in the stuffing-box, and then the oil-gland is set in place. A second packing is then inserted, and this is followed by a second oil-gland *X*. The gland *U* is then screwed on. The oil-gland is filled by means of a connecting line from the oil separator or a hand oil-pump *P*. There is usually a by-pass connection *R* from the hand-pump to the compressor cylinder, so that oil may be fed directly into the cylinder in case the piston packing-rings start to groaning. Garlock and Crandal packings are the favorite soft packings used on ammonia compressors; metallic packings with soft backing are also used with considerable success.

The piston-rod packing of a double-acting compressor requires greater attention than that of a single-acting machine. The wet system is particularly hard on packing, as there are few soft packings that will stand the freezing action of the liquid anhydrous ammonia without becoming hard and causing leaky stuffing-boxes. All piston rods should be packed carefully, but should not be packed at all if they are scored or in any way imperfect, as any imperfection on the rod will soon cut the packing. When possible, the rod should be ground perfectly circular, but if this is not possible, a very fine finishing cut should be taken, and the rod should then be thoroughly smoothed off with a fine grade of emery-cloth. Too much care can not be taken in packing the piston rods with metallic packing. No old rods should be thus packed without being thoroughly trued beforehand. Rods should be calipered at the end of the season, and any repairs on them should be made at that time. Care in this respect will soon pay for itself in the ammonia bill.

THE MOTOR.

1389. Speed of Compressor.—The maximum speed at which a compressor should run, even in the case of very small machines, is 100 revolutions per minute. If the compressor is speeded up above this point, the valves will not act quickly enough, and while the capacity is but slightly increased, the power required to drive the machine is directly in proportion to the number of revolutions. Compressors of capacities ranging from 10 to 25 tons should not exceed 70 revolutions per minute, while those of larger sizes and greater piston speed make from 50 to 60 revolutions per minute.

1390. Steam Engine.—Compressors operated by a steam engine as the motive power are usually direct connected with engines about as follows: Machines of 10 tons capacity or under, plain slide-valve engine; 10 to 25 tons inclusive, adjustable cut-off engine of the Meyer type; 30 tons and over, Corliss engine. A number of the very large machines with capacities ranging from 200 to 500 tons have in late years been provided with compound condensing engines. From this it will be seen that the relative efficiency of a small compression machine is much lower than that of the larger sizes. Owing to the low piston speed of the engine and consequent great amount of cylinder condensation, the steam consumption of the various classes is about as follows: Slide-valve engines, 50 pounds of steam per horsepower, 90 pounds of steam per hour per ton of refrigerating effect; Meyer cut-off, 40 pounds of steam, 65 pounds of steam per hour per ton of refrigerating effect; Corliss engine, 30 pounds of steam per horsepower, 50 pounds of steam per hour per ton of refrigerating effect.

The above are based upon a condensing pressure, or head pressure, of 150 pounds per square inch and a suction or back pressure of 15 pounds per square inch, both gauge pressures.

1391. Gas Engine. — Small compressors are often driven by gas or gasoline engines. Owing to the relatively high speed of the gas engine, it is necessary to belt such engines to the compressors. In case of gasoline engines, the economy is much better than that of a slide-valve engine, the cost of operating amounting to about 1 cent per horsepower per hour, or 2 cents per hour per ton of refrigerating effect. This is equivalent to coal at $2.50 per ton, not including attendance. For each ton of refrigerating effect, 2 to 2½ gas-engine horsepower should be allowed.

1392. Electric Motor. — Compressors driven by means of an electric motor are usually geared, rawhide pinions being used to decrease the noise. The motor should be of ample capacity to drive the compressor, so as to prevent the armature from burning out. 1½ to 2 kilowatts per ton of refrigerating effect is the usual practice.

1393. Water-Power.—Under high heads, the Pelton water-wheel is found to be the most efficient and best adapted water-wheel for driving ammonia compression machines. It is usually direct connected to the shaft, and is made with a large diameter so as to get a high circumferential velocity with 60 or 70 revolutions per minute. Under heads of less than 50 feet, a turbine is generally used. In most cases, it must be connected to the compressor by a belt or gearing which will reduce the speed to the proper number of revolutions.

1394. Position of Cranks.—The cranks of the compressor and direct-connected steam engine are set at 90° to each other. With this arrangement, the steam engine encounters its greatest resistance when the piston is at the middle of its stroke. If the cranks connected at 180°, or directly in line, the engine would encounter the greatest resistance when the piston is at the end of its stroke, and consequently has a low steam pressure to urge it forwards. Such a construction, though much cheaper, brings a much greater strain upon the cranks, shaft, and fly-wheels, and makes these latter parts do the work of compression with the energy stored in them. The fly-wheels should be well balanced and heavy. They should be carried on a shaft of ample size, and if possible out-board bearings should be provided to prevent deflection.

THE AMMONIA CONDENSER.

1395. Atmospheric Condensers.—The atmospheric condenser consists of a coil made up of straight pipes and return bends. The hot gas enters the coil at the top. Water is permitted to trickle over the coil in a thin film and condenses the gas within it, which after liquefying runs to the bottom of the coil and enters a receiver placed at that point. It will be seen that the hottest gas and the coldest water come in contact with the same part of the coil. This has a tendency to increase the head or condensing pressure, but it is necessary to let the gas in at the top in order that the liquefied gas will run to the bottom of the coil by gravity.

The De La Vergne condenser shown in Fig. 329 is constructed with a series of return bends *B* having pockets *P*.

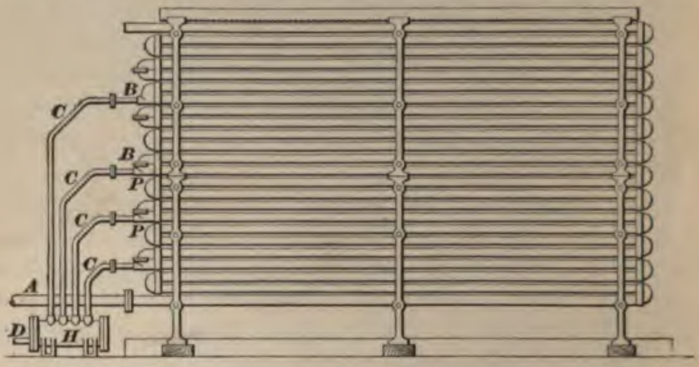


FIG. 329.

These pockets are connected to a header *H* at the bottom of the condenser by means of pipes *C*, *C*. The header in turn is connected by a pipe *D* with the receiver. By this arrangement, the hot gas is enabled to enter the bottom of the coil at *A* and gradually work up towards the top. The liquid anhydrous ammonia is trapped out by the pockets and does not run back to mix with the incoming hot gas. Atmospheric condensers are usually made of 1¼, 1½, or 2 inch pipe, the latter size being used in the De La Vergne condenser. The coils are about 20 feet long and 18 to 24 pipes high; 22 to 30 square feet of surface per ton is the usual allowance, depending upon the temperature of the condensing water, but 40 square feet will give more satisfactory results. The principal objection to atmospheric condensers is that they are least efficient when most needed. In hot weather, when the humidity is great, their action is very sluggish and the efficiency of the machine is diminished to a considerable degree. They are, however, low in first cost, easy to keep clean, and easily repaired.

Atmospheric condensers require from 3 to 4 gallons of water per minute per ton of refrigerating effect in hot summer weather, with water at 75° to 80°.

1396. Submerged Condensers. — The submerged condenser consists of a coil or coils of pipe submerged in a tank of water. The incoming gas enters at the top of the coil, and the outlet for the condensed liquor is at the bottom of the coil, the number of coils in the tank depending upon the size of pipe used and the capacity of machine. 1, 1¼, and 1½ inch pipe is used in these condensers, and 30 square feet of surface per ton is allowed in usual practice. The tanks should be deep rather than wide, so that the circulating water, which enters at the bottom of the tank and overflows at the top, will have an opportunity to reach a high temperature before leaving the tank. Condenser coils of 1¼-inch pipe should not exceed 350 feet in length. The inlets and outlets to all coils should be provided with valves or cocks, so that any coil may be turned off in case of leakage, without necessitating the shutting down of the plant. The outlets of the various coils are connected to a receiver, usually made of wrought iron with heads welded in. This receiver should have a capacity of about one-half gallon per ton of refrigerating effect, and should be provided with a pair of gauge-cocks and an air-valve for drawing off air or other impure gases. The water economy of a submerged condenser varies directly with the temperature of the water. With condensing water at 60°, 1½ gallons per ton is good practice, but 2 gallons should be used when the temperature is as high as 75°.

1397. The Surface Condenser. — The Hendrick surface condenser consists of a heavy cast-iron shell provided with water coils of extra heavy pipe. The action of this condenser is exactly the reverse of that of the submerged condenser. The water circulates through the coils; the gas enters the shell at the top, condenses, and falls to the bottom. The lower portion of the condenser forms the receiver of the anhydrous liquid and is provided with gauge-cocks and necessary fittings. On account of the large gas space and rapid water circulation, this condenser is very efficient; it gives a very low condensing pressure for any given water temperature and is also economical in the use of water.

GAUGES, VALVES, AND OTHER DETAILS.

1398. Pressure Gauges.—The pressure gauges used for ammonia are in construction similar to steam-pressure gauges except that the tube is made of steel for the reason that brass will not withstand the action of ammonia. The gauges register pressure only, or pressure and vacuum both, as may be desired. Cheap diaphragm gauges are made, but they are not accurate, and are seldom used by reputable builders.

1399. Liquid Level Gauges. — The anhydrous receiver and the oil trap should each be provided with a pair of liquid level gauges. These are of heavy cast iron or steel, and should be so arranged that they can be quickly closed in case of the glass breaking. Fig. 330 shows a section of the Hiller automatic safety gauge-cock, which is designed to shut off automatically in case of the glass breaking. It is also arranged so that the valve can be ground without taking the pressure off the vessel. The valve *A* is opened by a fork *D* attached to the valve-stem *B*. The stem *C* is used for grinding the valve.

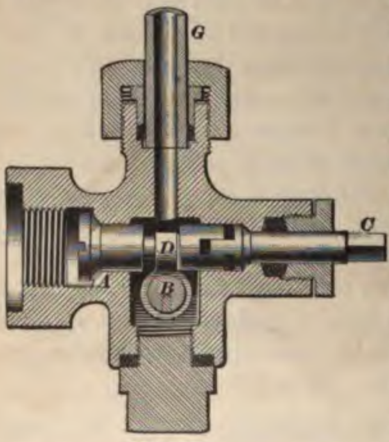


FIG. 330.

1400. Pipe.—All pipe used in ammonia work should be extra heavy, even for the low-pressure lines. Such pipe should be strictly of wrought iron, steel pipe being unsuitable for this purpose. In case any bent-pipe coils are used, they should be made of lap-welded, redrawn, or tuyère pipe.

1401. Fittings and Valves.—Fittings used on ammonia work are made of cast iron, malleable iron, or steel. They are protected as follows: (1) By means of a jam or lock-nut bearing against a rubber gasket which sets in a recess turned in the fitting; in case of a leak, a slight turn of the jam-nut forces the gasket well into the counterbore and thread and stops the leak. (2) By means of a gland fitting similar to the lock-nut, save that a gland is held against the gasket by means of bolts, which are fast to the flange of the fitting. (3) By means of either malleable iron or semisteel fittings, in which there is a slight recess or counterbore. After the pipe is made up, solder is run into the counterbore with a blowpipe, and a perfectly tight joint is made. Such joints are very satisfactory, except in lines that are subjected to both excessive heat and cold, in which case the solder is liable to crack.

Unions used in this work are flange unions with male and female joints, with lead packing in the groove.

Valves are of extra heavy pattern, all iron, and of either gate, globe, or angle type. The seats of such valves are usually soft metal, which can readily be repacked. Hard-metal valves with steel seats are not effective, as the liquid anhydrous ammonia appears to cut the steel.

Cocks used in this work are of the Fairbanks asbestos-packed pattern, the plug being packed with asbestos packing under a heavy hydraulic pressure. Cocks should be returned to the factory for repairs.

1402. Oil Trap and Strainer.—Between the compressor and condenser an oil trap is placed for the purpose of eliminating any oil that might pass over from the compressor cylinder with the ammonia gas. As shown in Fig. 331, it consists of a cylinder having an inlet *D* for the gas on the side about half way up from the bottom, and an outlet *B* at the top. The lower portion of the cylinder acts as an oil receiver, and is provided with a pair of gauge-cocks and a glass *C* for determining the quantity of oil in the receiver. Some builders place the inlet opening at the top with a pipe on the inside, leading the gas well down into the receiver, but such an arrangement is not as effective for eliminating the oil as the one above described, where the incoming gas impinges on a flat surface. The strainer consists of a large pair of flanges, with a piece of very fine perforated iron between them. The aggregate area of the perforations is greater than that of the pipe. The strainer is placed in the suction line and is intended to prevent any scale, dirt, or grit from entering the compressor cylinder. It should be cleaned at least once a season.

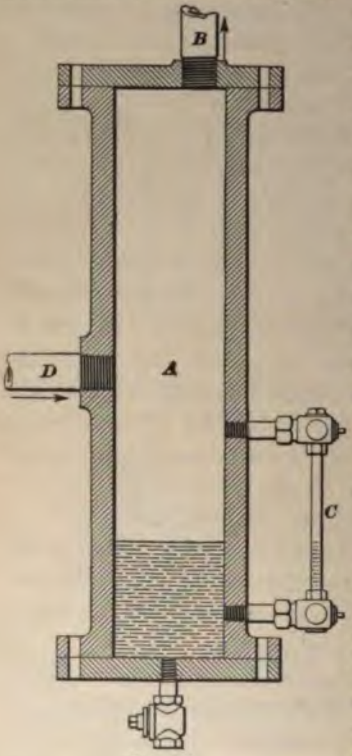


FIG. 331.

1403. Drier.—The best equipped compression machines are provided with **driers.** A drier consists of a vertical pipe 10 or 12 inches in diameter and 4 to 6 feet long, provided with removable heads. The shell is partly filled with quicklime or caustic soda and placed in the suction line of the compressor. The ammonia gas passing to the compressor must traverse the drier and there give up any moisture that it may contain.

1404. By-Pass.—It is often necessary, in making repairs, to empty the condenser into the expansion coil. For this purpose, the compressor is provided with a by-pass, which permits the ammonia to be taken from the condenser and delivered to the expansion coils, where it is stored until after the repairs are completed. A by-pass for a single-acting machine is shown in Fig. 332. When the condenser is to be pumped out, valves *A* and *B* are closed and valves *X* and *Y* are opened. To pump out the pipe *A C*, valves *A*, *X*, and *Y* are closed and valve *B* is opened.

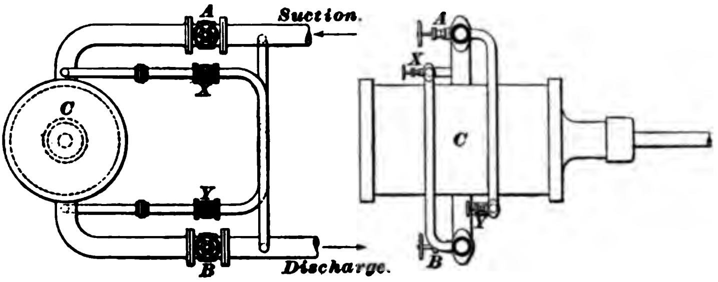


Fig. 332.

The by-pass on the Frick duplex compressor is shown in Fig. 333. The angle-valves *D* and *D'* are so arranged that when one of them, say *D'*, is open and the valve *C'* is shut, compressed gas will pass from the cylinder *B* through the discharge-pipe; when, however, *D'* is closed and *C'* is open, the gas will flow from *B* through the pipe *M*. The valves *S* and *S'* act in a similar manner. The valves *E* and *E'* are so arranged that when *E* is closed, the gas will flow through the pipe *M* from *C'* to the valve *G*, but when *E* is open, the gas will flow through the pipe *K* to the suction-pipe without passing to the valve *G*.

To pump out the compressor *B*, the valves *C'*, *G*, and *D* are opened and the valves *S*, *S'*, *D'*, *C*, *E*, *E'*, and *G'* are closed. The closing of the valve *S* shuts off the suction-pipe from the cylinder *A*, and the gas is pumped from cylinder *B* instead. It passes through the open valve *C'*, the pipe *M*, and the open valve *G*, and is expelled after compression into the discharge through the open valve *D*.

To pump out the condenser, the valves *D*, *C*, *G'*, *C'*, and *E* are opened and all others are closed. The valve *S'* being closed, the gas can not be drawn from the suction-pipe. As the piston in *B* makes its compression stroke, the gas will flow from the condenser through the discharge, the open valve *C*, the pipe *N*, the open valve *G'*, and thence into the cylinder *B*. The valve *D'* being closed, the compressed gas will flow through *M*, through the open valve *E*, and through

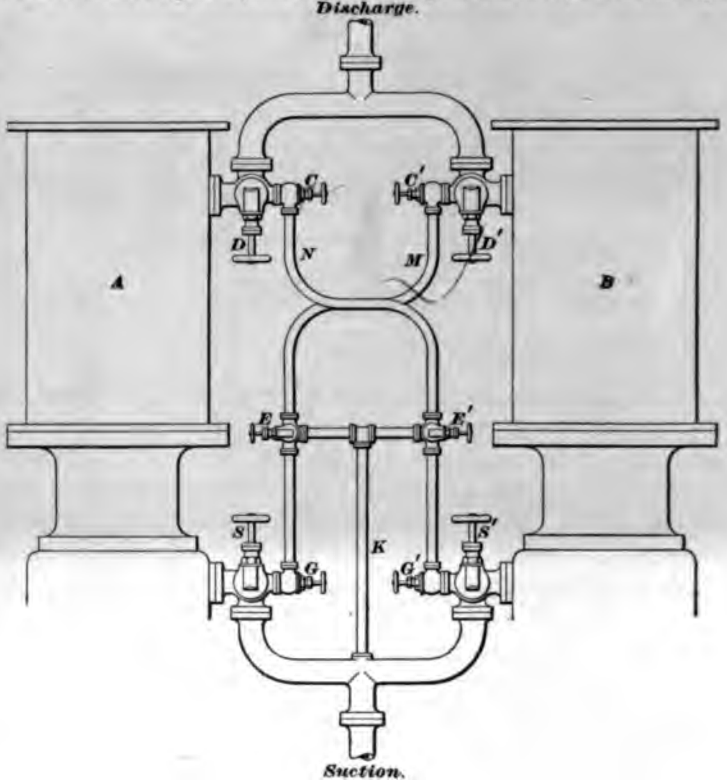


FIG. 333.

the pipe *K* into the suction line. With the valves thus arranged, pump *B* takes gas from the condenser and discharges it into the suction line. The pump *A* does no work whatever except to churn the gas back and forth.

CONSIDERATIONS AFFECTING THE ECONOMY OF THE COMPRESSION SYSTEM.

1405. Friction.—The total work delivered to the refrigerating-machine, including the steam engine or other motor that drives it, is the work done by the steam in the engine cylinder. The net or useful work is that done on the gas in the compressor cylinder. The difference between the two is the work done against the frictional resistances of the engine and compressor. To determine the friction, therefore, it is only necessary to determine by means of an indicator the indicated horsepower of the steam cylinder and that of the compressor cylinder; the difference will be the power absorbed in overcoming friction.

The ratio of the friction horsepower to the horsepower of the steam cylinder depends in some degree upon the type of machine. In this respect, the direct-acting has an advantage over the indirect machine; based on the horsepower of the steam cylinder, the friction of the former type varies from 12 to 20 per cent., while that of the latter type is from 18 to 30 per cent., the increase being due to the increased number of bearings and wearing surfaces. In general, the friction of a vertical machine is less than that of a horizontal machine.

1406. Clearance.—This very important factor in the economy of the compression system often cuts down the capacity and economy of the pump. In a first-class compressor, the clearance has been known to be as low as $\frac{3}{10}$ of 1 per cent. In actual practice, however, it probably amounts to about 1 per cent. The method of ascertaining the clearance space between the cylinder-head and piston has already been explained in Art. **1386.** That in the recess of the suction and discharge valves may be calculated as follows:

Rule.—*Multiply the area of the valve in inches by the distance between the face of the valve and the head when the valve is seated. Divide this product by the volume of the cylinder, and the result is the per cent. of clearance for the valve.* Applying this rule to each of the valves, and adding the results to the clearance space between the head and piston, the sum will be the total clearance for the compressor.

1407. Cylinder Superheating.—As stated in Art. **1380,** the greater the difference of temperature between the inlet and outlet gas, the greater the amount of cylinder superheating; therefore the warmer the condensing water and colder the brine, the more the vapor is superheated. For single-acting machines the loss due to cylinder superheating amounts to about 22 per cent. and for double-acting machines about 30 per cent. when the machine is running with a head pressure of 150 pounds and a back pressure of 15 pounds per square inch. If the head pressure is increased or the back pressure decreased, the loss will be greater.

1408. Compound Compression.—In order to overcome the loss due to cylinder superheating, compound ammonia compressors have been built. The gas is first compressed in a large cylinder and is then discharged into a smaller one, where it is compressed to the final pressure. There are, however, no authentic tests on such compressors, and it is an open question whether the increased friction does not more than make up for the gain due to the lessened superheating.

1409. Effect of Pressures.—Neglecting the loss due to superheating, it was shown in Arts. **1375** and **1376** that for maximum theoretical economy and capacity, the suction pressure should be as high as possible and the condensing pressure as low as possible.

It is evident also that these are the conditions that reduce the loss due to superheating. It follows, therefore, that the back pressure should be high, not only in order to increase the capacity, but also to reduce the superheating.

THE AMMONIA ABSORPTION SYSTEM.

GENERAL DESCRIPTION.

1410. The action of a refrigerating system of the absorption type is based upon the affinity of a vapor, usually ammonia vapor, for water. As stated in Arts. **1361** and **1363**, water will absorb 1,140 times its volume of ammonia gas, and to liberate one pound of ammonia gas thus absorbed requires the expenditure of 925.7 B.T.U. Suppose we have a strong solution of aqua ammonia; if heat be applied to the solution, the ammonia gas or vapor will be driven off at relatively high pressure, and when passed through a condensing coil, will condense to the liquid state. The liquid can now, just as in the compression system, be admitted to a refrigerating coil through an expansion, valve. In this coil it will vaporize and withdraw heat from the surrounding objects. The vapor may now be again absorbed by water, thus regaining its original state and closing the cycle of operations.

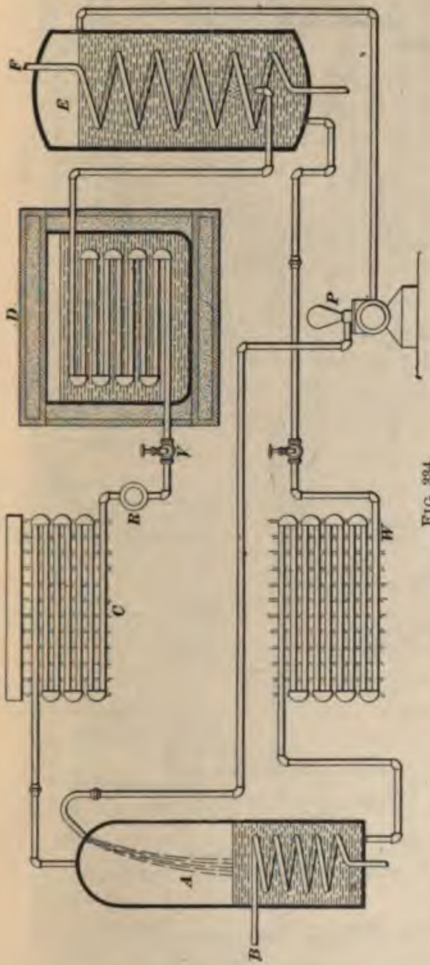


FIG. 334.

1411. The essential features of the absorption system are shown in Fig. 334. Steam is admitted at a pressure of about 40 pounds per square inch, gauge, to a coil *B*, submerged in a strong solution of aqua ammonia contained in the vessel *A*. The temperature of the solution will be raised nearly to that of the incoming steam, say to about 270° F., and the heat absorbed will cause the ammonia gas to be driven off

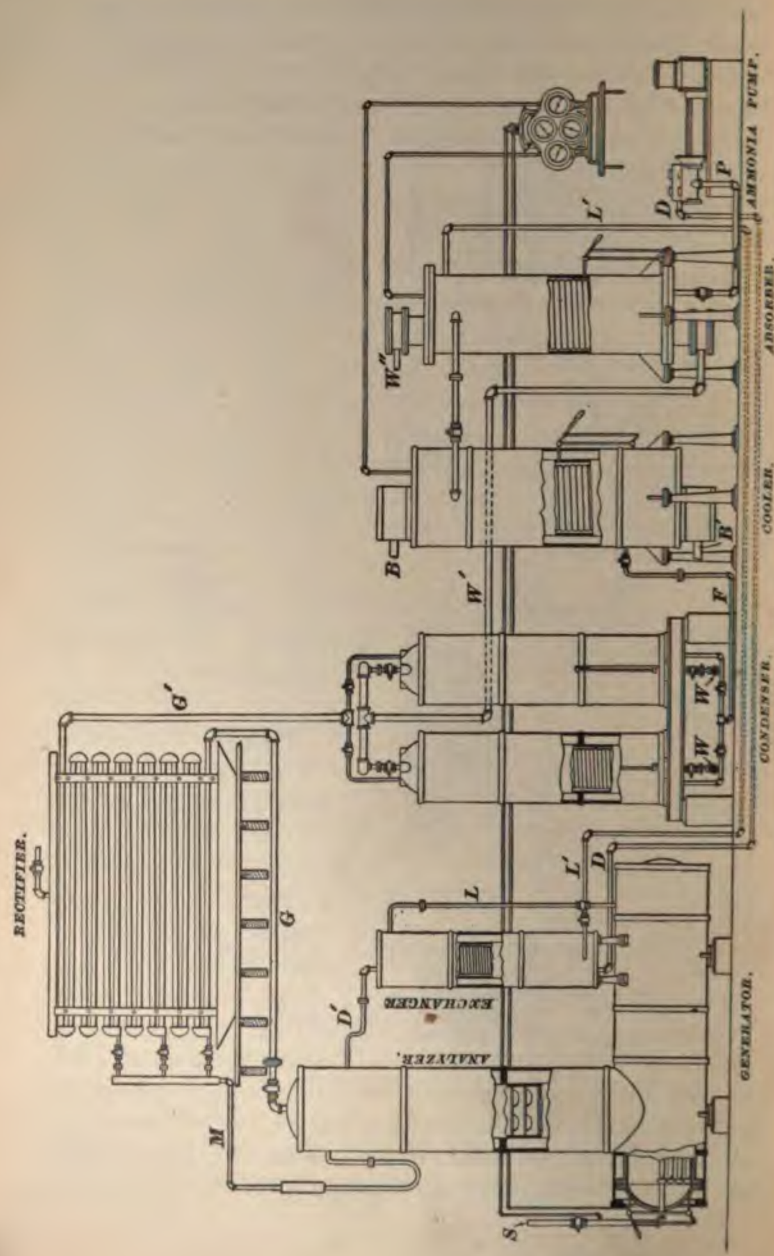

RECTIFIER.
G'
G
M
S
ANALYZER.
EXCHANGER
D'
L
L'
D
GENERATOR.
W
W
CONDENSER.
F
B'
B
W'
COOLER.
W''
ABSORBER.
L'
D
P
AMMONIA PUMP.

FIG. 335.

at a pressure of say 160 pounds per square inch. The pressure corresponding to any given temperature depends upon the strength of the solution; the stronger the solution the higher the temperature, and *vice versa*. As the temperature of the solution is below the boiling point of water for this pressure, no water will evaporate, and only ammonia gas will pass over into the condenser *C*. The construction of the condenser is similar to that of the compression system. The cold water flowing over the condensing coils absorbs heat from the gas, and the combined effect of the high pressure and the cooling action of the water is to liquefy the gas. It is to be noted that the pressure is not produced mechanically, as in the compression machine, but by chemical action.

As the ammonia liquid passes through the expansion-valve to the expansion coils, the pressure is reduced, reevaporation begins in the expansion coils, and heat is absorbed from the brine or other substance in the chamber *D*.

In the compression system, the ammonia pump draws the gas from the expansion coils, but in the absorption system, the removal of the gas is effected by allowing the gas from the expansion coils to mingle with the weak solution of ammonia from which the gas was expelled in the **still** or **generator** *A*.

During the process of generating the ammonia gas in the still *A*, the strong solution rises to the top on account of its smaller specific gravity, and the weaker solution settles to the bottom and flows through a pipe to the vessel *E* called the **absorber.** Here it meets the gas as it comes from the expansion coil and absorbs it. Since a low temperature is required for efficient absorption, the weak liquor on its way to the absorber passes through a coil *W*, which is cooled by running water. The absorber *E* is also provided with a water coil *F*. A small pump *P* takes the strong liquid from near the top of the absorber and forces it back into the generator. This completes the cycle of operations.

1412. Fig. 335 shows a sectional elevation of a Pontifex-Hendrick absorption machine. The various parts and their functions are as follows: The **generator, still,** or **retort,** is the cylinder containing the charge of aqua ammonia; it is supplied with a steam coil to drive the ammonia gas from the solution. The steam inlet is shown at *S*. The ascending gas passes through the trays of the **analyzer,** which act as baffle-plates, and cool the gas by means of the incoming strong liquor which they contain. From the analyzer the gas passes through the pipe *G* to the **rectifier;** this is a coil of the atmospheric type, supplied with pockets on some of the return bends to catch any moisture that may pass from the generator and be mixed with the ammonia gas. The gas on leaving the rectifier is strictly anhydrous, all water vapor having been removed. The water returns to the analyzer through the drip *M*. The gas then enters the **condenser,** where it is liquefied, then passes to the **cooler,** where, after abstracting the heat from the brine, it is revaporized and passes off to the **absorber.** The weak liquor in the generator is forced by the high pressure in the generator into the shell of the **exchanger,** whence it passes to the absorber. After absorbing the gas and being cooled by the water coils in the absorber, the liquor is forced by means of the ammonia pump through the coils of the exchanger and into the analyzer. The hot weak liquor entering the shell and the cool strong liquor entering the coil of the **exchanger,** transfer or exchange their heat, so that the weak liquor becomes cooled and the strong liquor becomes heated. The use of the exchanger effects a considerable saving of both water and coal. The strong liquor entering the analyzer gradually works down from tray to tray, and in its descent is heated by the ascending gas.

The various parts indicated by the reference letters are as follows:

B, brine inlet to cooler.
B', brine outlet from cooler.
S, steam inlet.
W, *W*, water inlet to condenser.
W', pipe conveying water from condenser to absorber.

W'', waste-water outlet from absorber.

L, pipe conveying weak liquor from generator to exchanger.

L', pipe conveying weak liquor from exchanger to absorber.

F, pipe conveying gas from condenser to cooler.

G, pipe conveying gas from analyzer to rectifier.

G', pipe conveying gas from rectifier to condenser.

D, pipe conveying strong liquor from pump to exchanger.

D', pipe conveying strong liquor from exchanger to analyzer.

P, pipe conveying strong liquor from absorber to pump.

DETAILS OF THE ABSORPTION SYSTEM.

1413. The **generator,** sometimes called the still or retort, is the vessel in which the aqua ammonia is placed and from which the ammonia gas is evaporated. There are two classes of generators, viz., direct fired and those indirectly heated by means of a steam coil. The former class is now seldom seen. Generators provided with steam coils are subdivided into two classes, viz., vertical and horizontal. Vertical stills, used largely on absorption machines in the South, are usually made of boiler plate with riveted seams. The internal coils are of the helical type nested one inside of the other. The steam inlets pass through the side of the shell, and the outlet for the condensed steam through the bottom head. These tails are protected by either lock-nuts or glands. The same objection applies to the vertical still as to the vertical boiler; it does not make dry gas. The evaporating surface being comparatively small, depending upon the diameter of the shell, the boiling is very rapid, and it is difficult to keep the steam coils always submerged. If these coils are exposed to the action of the gas and liquor when uncovered, they will become pitted and will have to be renewed in the course of two or three years. Vertical stills are also liable to boil over when forced, the ebullition being so strong at such a time as to cause the complete charge to pass over into the condenser. Another objection is the excessive height necessary to accommodate stills of this class.

With a view to the elimination of these difficulties, the horizontal still was constructed. The ebullition in this case is very quiet, no action being noticeable in a gauge-glass. The liberating surface of the liquid is large, and there is no difficulty in keeping the coils covered with liquor. Coils in use in this type of machine have lasted for fourteen years without any appreciable pitting. The shells of horizontal generators are usually made of cast iron. The first cost of a horizontal generator is greater than that of the vertical still.

1414. Analyzer.—In the case of the vertical still, the upper portion of the shell is provided with a series of wrought-iron plates so arranged that the returning strong liquor will pass over these plates and come in contact with the ascending gas. In the case of the horizontal still, the analyzer is attached as shown in Fig. 335. It is provided with cast-iron pans, one set above the other in such a manner that though the descending strong liquor comes in contact with the gas, the latter does not pass through the liquor as in the former case; the gas is thereby left much drier. Cast-iron plates are also more durable than the wrought-iron ones.

1415. Rectifier.—The rectifier is a coil or series of coils arranged like those in the condenser. Its office is to precipitate on the sides of the pipes any moisture that may have passed over with the gas. This moisture is then collected in a trap, like water in an ordinary steam separator, and the liquid is then allowed to pass back into the analyzer by gravity or enter the receiver of the absorber. The rectifier is a very necessary part of the absorption machine, as upon it the efficiency of this type of machine largely depends.

1416. Exchanger and Weak-Liquor Cooler.—The exchanger is a cast or wrought iron shell provided with coils. The inlet for the hot, weak liquor from the still enters the shell near the top, and the outlet is near the bottom. The ammonia pump delivers the strong liquor into the bottom of the coils. After being heated, the strong liquor passes out of the top of the coils into the analyzer. The inlets and outlets of these coils are connected by means of a header. Some makers reverse this action and allow the weak liquor to pass through the coils and the strong liquor through the shell. One method has no particular advantage over the other. The tails of the coils are provided with either lock-nuts or glands, to prevent ammonia leakage. Well-proportioned machines should have at least 6 square feet of exchanging surface per ton of refrigerating effect. Machines employing vertical stills seldom have over 2 square feet per ton, as better exchanging increases the liability of boiling over. In such machines, it is necessary to provide other means for cooling the weak liquor before it enters the absorber, as the surface is not sufficiently large in the absorber to cool the weak liquor below 130° or 140° F. For this purpose, a *weak-liquor cooler* is provided. It consists of either an atmospheric or submerged coil of pipe through which the weak ammonia liquor passes on its way from the exchanger to the absorber. The surface depends largely upon how much it is desired to cool the liquor, and may be from 3 to 10 square feet per ton.

1417. Condenser.—Condensers used for this system are similar to those already described in case of the compression machine in Arts. **1395** to **1397.**

1418. Absorber.—There are four classes of absorbers, viz., the *submerged* or *tank* absorber, the *full* absorber, *empty* absorber, and *atmospheric* absorber. The first consists of a series of coils arranged in a water-tank. The gas enters the top of the coils, and the weak liquor sprayed in at the top gradually works down through the coil, absorbs the gas in its descent, and enters the receiver at the bottom, into which all the coils are connected. The water in the tank enters at the bottom, passes off at the top, and carries away the heat of absorption. Such absorbers should contain from 30 to 40 square feet of surface per ton of refrigerating effect. The *full* absorber consists of a cylindrical shell provided with water coils nested one within the other, or else with straight tubes similar to the tubes of a boiler. The latter is preferable on account of ease of cleaning, but the liability of leakage is greater, owing to the number of joints. The gas and weak liquor enter the shell near the bottom, the gas passing up through the liquor contained in the shell and being absorbed by it. The liquor overflows near the top of the shell and passes to the pump suction, the water entering the top of the coils or tubes and out at the bottom. The objection to this type is that it is necessary for the gas entering the absorber to have sufficient pressure to pass through the liquor. This takes away from the absorber the pressure equal to the head due to the height of liquor above the inlet gas. For this reason, the *empty* absorber is much more efficient, particularly in the case of low temperatures. The construction of this absorber is similar to that of the full absorber, except that the weak liquor is sprayed in over the coils at the top of the shell and the gas enters below this spray. The bottom of the absorber takes the place of the receiver. The *atmospheric* absorber is not a commercial success, and is only provided by one or two builders.

1419. Ammonia Pump.—The aqua-ammonia pump consists of a small well-proportioned steam or power pump, amply strong to run against a pressure at 350 to 400 pounds per square inch. This is not the regular service of the pump, but such pressure is used for the purpose of testing the various parts of an absorption machine and the piping for the purpose of detecting leaks before the machine is started. Direct-acting steam-pumps are built either with or without fly-wheels. Duplex ammonia pumps are not satisfactory. Ammonia pumps should have a capacity of 150 cubic inches per minute per ton of refrigerating effect.

1420. Castings and Coils.—All castings used for ammonia work should be made in either loam or dry sand and should be tested to a hydraulic pressure of 500 pounds per square inch. The iron used should be of fine grain and free from imperfections, sand, or blow-holes. All coils used for this work should be made of extra heavy lap-welded redrawn tuyère pipe. They should be continuously welded and tested to a hydraulic pressure of 800 pounds per square inch, or an air-pressure of 300 pounds per square inch under water. The coils should be well hammered while being tested, so as to dislodge all scale, dirt, etc.

CONSIDERATIONS AFFECTING THE ECONOMY OF THE ABSORPTION SYSTEM.

1421. Generation of Anhydrous Ammonia.—Few of the absorption machines now built are able to generate pure anhydrous ammonia gas without some trace of moisture. In order to get economical results, the absorption machine must make anhydrous ammonia. Any adulteration, even if as low as 5 per cent., decreases the economy of the machine considerably. Machines having vertical stills are more subject to this trouble than those with horizontal stills. The analyzer of a horizontal still condenses out some of the water on the under side of the analyzer trays, the gas leaving with probably not over 5 per cent. entrained moisture. To extract this remaining 5 per cent., the rectifier is used. The most efficient form of rectifier is that containing a number of drip pockets for the purpose of trapping out the moisture by a differential process; the gas leaves the last pocket in a practically dry, anhydrous state.

The method of testing samples of anhydrous ammonia for impurities has already been explained in Art. **1364.** If it is not convenient to draw out a sample of anhydrous ammonia, its condition may be approximately determined by noting the temperature of the brine, or, better yet, that of the ammonia in the expansion coils, if this is possible, and the back pressure. After determining these two readings, turn to Table 26, giving the properties of saturated ammonia gas. The temperature of the ammonia due to the back pressure as given by the table should be nearly identical with the temperature in the expansion coils. If the latter temperature is any higher than that given in the table for the corresponding pressure, one may be assured that the variation is due to water in the anhydrous ammonia. The ice-machines of the South are much troubled in this particular, as few of them carry a back pressure above that of the atmosphere, even with high brine temperatures in the freezing tank. This is due to the fact that the machines do not make anhydrous ammonia. The makers and operators of such machines think that it is necessary to carry a low back pressure; this is a mistake, and the trouble should be looked for in the still, analyzer, or rectifier of the machine; when these are constructed correctly, the back pressure will rise of its own accord.

1422. Efficiency of Absorber. — Naturally the higher the back pressure, the stronger will be the strong liquor leaving the absorber; and the stronger the liquor, the less quantity will have to be pumped per ton of work done. From this it will be seen that a high back pressure is conducive to economy, in that it tends to secure a strong solution in the absorber and thus diminishes the quantity of liquor to be pumped. The absorber that will give the strongest solution for any given back pressure and the coolest pump delivery for the same amount of water is the most efficient. Absorbers run on the empty principle are more efficient in this respect than either the submerged or full absorber. An absorber should have sufficient surface and so distributed that it will be economical in the quantity of cooling water used. It should be designed with a view to using the water after it has passed through the condenser and has done its work in condensing the ammonia gas. Water should never be run through the absorber first and then through the condenser, as this invariably increases the head pressure.

1423. Efficiency of Exchanger.—The exchanger should be of ample size and have large surfaces, so that it is unnecessary to add on a weak-liquor cooler. A large exchanger naturally gives cool weak liquor to the absorber and warm strong liquor to the analyzer. This economizes both the steam and water consumption of the machine.

1424. Comparison of Absorption and Compression Systems.—Recent tests made by Professor J. E. Denton on a 50-ton Pontifex-Hendrick machine have demonstrated the following facts: That with excessively low brine temperatures, such as 20° below zero F., the capacity of the machine was only diminished by about 10 per cent. The economy of the machine at these low temperatures was better than that of a compression machine driven by a compound Corliss engine that would not use over 15 pounds of steam per horsepower per hour. Its economy with high brine temperatures was better than that of a compression machine driven by an ordinary Corliss engine. The water consumption of this machine with low brine temperatures was from two to three gallons per ton per minute, but the temperature of the water in this case was 70° F. This is probably considerably more than would be used by a compression machine.

The relative economy of the two systems is still an open question. Under certain conditions, it is likely that the absorption machines give better results than compression machines; for example, a small absorption machine will probably be more economical than a compression machine driven by an ordinary slide-valve engine. On the other hand, tests of large compression machines under favorable conditions show an economy higher than that of the ordinary absorption system of the same capacity.

CARBON-DIOXIDE REFRIGERATING-MACHINES.

1425. Compression machines using carbonic acid as a refrigerating fluid are coming into quite extensive use, especially in Germany, where liquid carbonic acid is manufactured very cheaply as a by-product of the brewing industry.

The characteristic feature of the carbon-dioxide system is the high pressure required for the condensation of the gas. Referring to Table 25, Art. **1355,** it is seen that at a temperature of 68° F., the pressure is 864 pounds per square inch, absolute. The back pressure is correspondingly high; for a temperature of 5° F. in the expansion coil, the pressure is 342 pounds, absolute.

1426. Reference to Table 25 shows also that at ordinary refrigerating temperatures, the density of carbon dioxide is great compared with ammonia at the same temperature. Thus, at 77° F., a cubic foot of carbon dioxide (vapor) weighs 15.475 pounds, while a cubic foot of ammonia at that pressure weighs only .5 pound. Though the latent heat of carbon dioxide—and in consequence the refrigerating effect per pound—is less than that of ammonia, the greater density permits the use of a compression cylinder of much smaller dimensions than the cylinder of the ammonia compressor. This may be illustrated by a numerical example. In the problem, Art. **1374,** the theoretical volume of the compressor cylinder for a capacity of 20 tons was found to be a little less than ½ cubic foot. Suppose we have a compressor using carbon dioxide, the condenser temperature being 68° and the temperature in the refrigerating coil 14° F. The heat abstracted *per pound* of vapor is given by equation (c), Art. **1370:**

$$Q_2 = r - s\,(t_c - t_a).$$

The specific heat of liquid carbon dioxide being 1, we have

$$Q_2 = r - (t_c - t_a).$$

At 14° F., the latent heat is 115.7 B. T. U. (see Table 25); hence

$$Q_2 = 115.7 - (68 - 14) = 61.7 \text{ B. T. U.}$$

For a capacity of 20 tons, the heat abstracted per minute must be

$$\frac{20 \times 285{,}300}{24 \times 60} = 3{,}962.5 \text{ B. T. U.}$$

At 112 strokes per minute, the weight of carbonic acid circulated per stroke must be

$$\frac{3,962.5}{61.7 \times 112} = .57 \text{ lb.}$$

At a temperature of 14° F., a cubic foot of carbon dioxide weighs 4.535 lb.; hence the cylinder volume required is $.57 \div 4.535 = \frac{1}{8}$ cubic foot, about $\frac{1}{4}$ of the volume required for ammonia.

1427. The small volume of the compressor is an undoubted advantage of the carbon-dioxide machine. Other advantages claimed are the following:

Carbon dioxide, or carbonic acid, is cheaper than ammonia, is non-corrosive, non-explosive, and is not dangerous to life when diluted with air. Another advantage of carbon dioxide is that it has no harmful effect on beer, meat, water, or any articles in cold storage.

The objections to the carbon-dioxide machine are as follows: The high pressures employed render the construction and operation of the machine difficult. It is also difficult to construct the piping and coils so that the joints will withstand the pressure. Gauge-glasses will not stand the pressures. Owing to the small size of the compressor, the prejudicial effect of the clearance is more marked in the carbon-dioxide machine. Carbon dioxide being colorless and odorless, it is difficult to detect leaks. The usual test for leaks is lime-water, which becomes milky in the presence of carbon dioxide.

1428. In the expression for the heat abstracted,

$$Q_2 = r - s\,(t_c - t_a),$$

the subtractive part, $s\,(t_c - t_a)$, is due to the incompleteness of the cycle (see Art. **1369**). The latent heat of ammonia being relatively great, 500 B. T. U. and more per pound at ordinary temperatures, the loss $s\,(t_c - t_a)$ due to fall of the temperature of the liquid is small in comparison. In the case of carbon dioxide, however, the latent heat is much less, and this loss is relatively much greater; thus, in the example, Art. **1426**, the loss, 68 − 14 = 54, is nearly one-half of the latent heat, 115.7.

To decrease the loss of efficiency due to the incompleteness of the cycle, it has been proposed to introduce a motor between the condenser and refrigerator, upon which the expanding liquid will do work, thus rendering the cycle complete.

With an expansion-valve, the efficiency of a carbon-dioxide machine is about two-thirds that of an ammonia compression machine.

SULPHUR-DIOXIDE REFRIGERATING-MACHINES.

1429. Refrigerating-machines using sulphur dioxide as a refrigerating fluid are in use to some extent. However, a reference to Table 24, giving the properties of saturated sulphur dioxide, shows that this gas is not especially adapted for a refrigerating fluid. Its latent heat per pound is less than $\frac{1}{3}$ of the latent heat of ammonia, while its density is but little greater than that of ammonia. Hence, for the same ice-making capacity, the volume of the compressor cylinder must be about three times the volume of an ammonia compression cylinder. This naturally increases the first cost of the plant, and even allowing that such a machine is as economical as an ammonia machine, its first cost is sufficient to debar its sale. Another serious objection to the sulphur-dioxide machine is the fact that the utmost care must be taken that no gas shall escape from the condenser and other coils, as this gas combining with water forms sulphurous acid, which rapidly attacks iron. The machines first built had copper condensing coils, but the first cost of these was so great that they were soon abandoned. The utmost caution must be exercised in keeping the piston rod perfectly dry, to prevent corrosion. Sulphur dioxide is poisonous when inhaled.

For low refrigerating temperatures, the pressure in the refrigerating coils is less than atmospheric pressure (see

Table 24). As a result, air is likely to leak into the coils, and this in combination with sulphur dioxide will form sulphur trioxide, which in turn will in contact with water form sulphuric acid.

At very low refrigerating temperatures, the efficiency of the sulphur-dioxide machine is low; at higher temperatures, the efficiency compares fairly well with that of the ammonia compression machine.

VACUUM REFRIGERATING-MACHINES.

1430. The vacuum machine employs water vapor as the refrigerating fluid. From the fact that water vapor in order to have a low temperature must have a very low tension arises the name "vacuum" machine.

The operation of the vacuum machine is precisely similar to that of an ammonia compression machine. The vacuum is formed by a pump, which withdraws the vapor from the refrigerator, where the pressure is about .1 pound per square inch or less, and compresses it into a condenser at a pressure of about 1.5 pounds. The evaporation of a part of the water in the refrigerator withdraws enough heat from the remainder to turn it to ice; or, if the refrigerator contains brine, the heat absorbed by evaporation lowers the temperature of the brine.

In a vacuum machine of this type, the vapor cylinder must have a capacity of about 150 times that of an ammonia compression machine for the same tonnage. The number of gallons of condensing water per ton of ice-melting capacity, assuming a range of 30° F. in the condensing water, is 340. The ice-melting capacity per pound of coal, assuming three pounds of coal per hour per horsepower, is about 25 pounds.

It is evident that the enormous size of a vacuum machine of this type puts it out of competition with other refrigerating-machines.

1431. In another form of vacuum machine, the use of the large compressor is avoided by the use of sulphuric acid as an absorbent. Fig. 336 is a diagram showing the principle of operation of this machine. By means of the air-pump *P*, a nearly perfect vacuum is produced in the chamber *A*, and the water in the bottom of this chamber begins to vaporize. The vessel *B* contains sulphuric acid, which is delivered to the vessel *C* in the form of a spray. The acid having a great affinity for water absorbs the vapor in the chamber, and the dilute acid flows out into the vessel *D*. Fresh water enters the chamber through the water injector *F*, and carries with it some of the salt solution that has been passing through the refrigerating coil *E*. The vaporization of some of the water chills the remainder, which falls to the bottom of the chamber, and passing to the coil *E*, absorbs a fresh supply of heat. A pump, not shown in the figure, is required to pump the acid after it is reconcentrated into the vessel *B*.

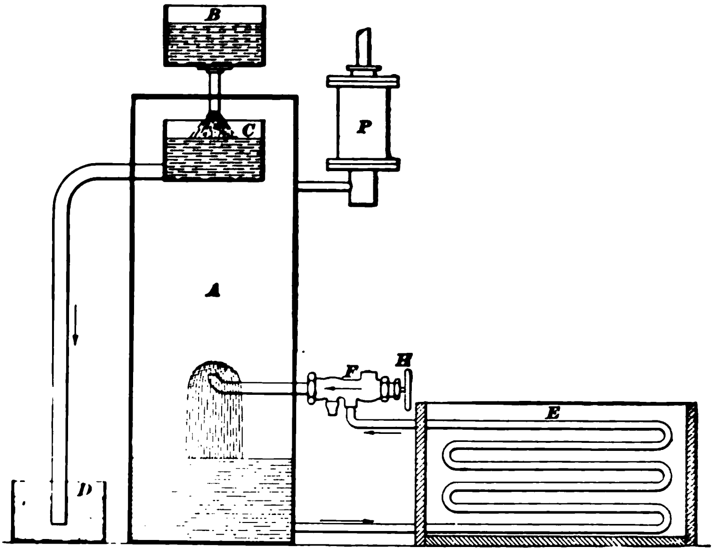


FIG. 336.

The principal objection to the machine just described is that the sulphuric acid is an inconvenient liquid to handle. The vessels and pipes containing it must be of lead or be lead-lined.

APPLICATION OF REFRIGERATION.

REFRIGERATING SYSTEMS.

1432. There are two principal systems of refrigeration, viz., the **brine** system and the **direct-expansion** system.

In the former system, the expansion coils (*B*, Fig. 324) are immersed in a tank of brine; this brine, which is a non-freezing solution of salt, gives up its heat to the ammonia evaporating in the coils, and is then pumped through coils of pipe placed on the sides or ceiling of the room to be cooled. The circulating brine thus continually absorbs heat from the cold room and gives it to the ammonia, and the latter carries it to the condenser.

1433. In the direct-expansion system, the ammonia is admitted through the expansion-valve directly into the coils in the rooms to be refrigerated. The heat of the cold room is taken up by the ammonia directly, and the intermediate agent, brine, is not employed. The difference between the two systems may be explained as follows: In Fig. 324, suppose the expansion coil *B* to be a comparatively short or compact coil, and let the vessel *D* be a tank containing brine; with this arrangement, we have the brine system. On the other hand, suppose the vessel *D* to represent the room or rooms to be cooled, and suppose the expansion coil *B* to be a long coil, divided into many branches, and located on the ceilings or sides of the rooms; this arrangement constitutes the direct-expansion system.

1434. **Advantages of the Systems.**—Each of the systems just described has certain advantageous features, which are, however, dependent to some extent on the conditions under which the system is operated. Advocates of the brine system urge the following points in favor of this system and against the direct-expansion system:

1. The total weight of ammonia required by the brine system is less than that required by the other system.

2. The whole of the ammonia part of the system is

located in one room, under the direct care of the attendant; with the direct-expansion system, on the other hand, the close attention of the engineer is required in every room.

3. On account of the large quantity of circulating brine, the temperatures are more easily regulated with the brine system than with the other; also, the brine will circulate several hours after the machine has stopped and still keep the rooms cold. In the direct-expansion system, this reservoir of cold—as the brine may be called—is lacking, and as soon as the machine stops, the refrigeration also stops, and the temperature of the rooms begins to rise. With direct expansion, therefore, it is necessary to keep the machine running continually 24 hours per day, while with the brine system the machine can, if desired, be shut down at night and started the next morning. This is a decided advantage for the brine system in the case of small plants, where the work required is not sufficient to justify the employment of a night attendant. In large systems, also, where the machines run continuously, it is of some advantage to be able to stop the machine for some hours to make needed repairs.

4. In cold-storage refrigeration, the danger of damaging goods by a leaky ammonia coil is a serious objection to the direct-expansion system.

The points advanced by those who favor the direct-expansion system are as follows:

1. The system is simpler and cheaper to install; there being no intermediate agent to circulate, the brine tank, expansion coils in the tank, and brine-pump are dispensed with.

2. The economy of the direct-expansion system is superior to that of the brine system. In the first place, the power required to drive the brine-pump is saved. Then the temperature of the ammonia in the expansion coil must be lower in the brine system than in the direct-expansion system. Suppose, for example, that the cold room is to be kept at a temperature of 32° F. By using a sufficient amount of piping, the temperature of the ammonia in the expansion coils may be say 22°, the drop of 10° being necessary

to effect the transfer of heat. The back pressure corresponding to 22° is about 35 pounds gauge. If brine is used at a temperature of 22°, it is evident that the temperature of the ammonia in the expansion coils must be lower in order that the brine may give up its heat to the ammonia. Allowing a drop of 10°, the temperature in the expansion coils is 12° and the corresponding suction pressure is about 25 pounds per square inch, gauge. It was shown in Art. **1375** that the capacity and economy of a machine depend very largely on the suction pressure; hence, since with the brine system the suction pressure is necessarily lower than with the direct-expansion system, it follows that the latter system is the more economical, other things being equal.

The brine system is generally preferred for small installations and the refrigeration of cold-storage boxes in markets, hotels, etc., where the large number of boxes would require too close attention with direct expansion. For large installations, the superior economy of the direct-expansion system is a strong point in its favor.

THE DIRECT-EXPANSION SYSTEM.

1435. The pipe used in direct-expansion coils varies from 1 inch to 2 inches in diameter. The coils are preferably made of continuously welded, extra heavy pipe, but this is not possible in the case of 2-inch pipe coils when made up with steel return bends and soldered joints. The length of coils used depends entirely upon the builder, as there are no regular rules or formulas for this work. About 400 feet for 1-inch and 1¼-inch pipe and 600 feet for 1½-inch and 2-inch pipe is a maximum. Rules for piping for direct expansion are entirely empirical. The following is the average practice:

High-temperature work, such as brewery refrigeration, packing-house work, and all work above 32° F., 1 lineal foot of 1¼-inch pipe to every 16 cubic feet of space. For freezing rooms to be held at a temperature of 15° F. or under, 1 foot of 1¼-inch direct-expansion pipe for 6 cubic feet of space.

Pipes of larger diameter will carry more space in proportion to their diameter. The quantity of piping thus specified is based upon a back pressure of 15 pounds per square inch, gauge, which is equivalent to a temperature of 0° F. If the back pressure is 30 pounds, double the quantity of pipe should be used, and if as low as atmospheric pressure, one-half the quantity is all that is necessary. Flat metal disks are at times attached to direct-expansion coils for the purpose of increasing the radiating surface. Opinions differ as to the efficiency of these disks. Details of the piping of cold-storage warehouses, breweries, etc., will be given in a subsequent section.

All coils should be provided at the inlet with expansion-valves, that is, valves for regulating the amount of anhydrous ammonia entering the coil, sometimes known as *feed* valves. The outlet valves are the full size of the coil and act as stop-valves for shutting it off.

The feed and return lines should have the following sizes:

Capacity.	Main Feed Line.	Main Return Line.
5 tons.	½ inch.	1¼ inches.
10 "	½ "	1½ "
25 "	½ "	2 "
40 "	¾ "	2½ "
75 "	1 "	3 "
120 "	1¼ "	4 "

The above table is for the maximum tonnage that can be secured with any given size of pipe, the velocity of the gas being 80 feet per second. In case of return lines of any great length, say over 100 feet, a pipe one size larger should be used. All direct-expansion coils should be provided with drip pans for catching the melted frost.

THE BRINE SYSTEM.

VARIETIES AND PROPERTIES OF BRINE.

1436. Salt Brine.—There are two salts commonly used for making the brine used in brine circulation. The first is Liverpool salt (chloride of sodium), which forms the ordinary brine capable of withstanding a temperature of about 0° F. This salt is cheap in first cost, but has a corrosive action on iron.

The following table gives the percentage by weight of the salt in a brine solution of a given hydrometer reading, and the freezing point of that solution:

TABLE 28.

PROPERTIES OF SALT BRINE.

Degrees Beaumé, 60° Fah.	Degrees on Salometer 60° Fah.	Specific Gravity, 60° Fah.	Per Cent. of Salt by Weight.	Weight of One Gallon.	Weight of One Cubic Foot.	Freezing Point, Degrees Fah.	Specific Heat.
0	0	1.000	0	8.35	62.40	32.00	1.000
1	4	1.007	1	8.40	62.80	31.80	0.992
5	20	1.037	5	8.65	64.70	25.40	0.960
10	40	1.073	10	8.95	66.95	18.60	0.892
15	60	1.115	15	9.30	69.57	12.20	0.855
19	80	1.150	20	9.60	71.76	6.86	0.829
23	100	1.191	25	9.94	74.26	1.00	0.783

Care should always be exercised in selecting a good salt for making brine. Ordinary desert salt will not answer for this purpose, as it will not give a strong enough solution on account of being impure. Such salt always gives trouble by having ice form on the expansion coils, and in case of ice-making, by freezing the cans fast to the coils.

1437. Chloride of Calcium Brine.—The other salt used for making brine is the chloride of calcium. It has all the most essential properties which ordinary salt lacks. It has no corrosive action on iron, which makes it unnecessary to have the brine-pump lined with brass. It has, in fact, an oily nature, and for that reason has a strong tendency to leak if the piping is in any way imperfect; care should therefore be exercised in the pipework of a chloride of calcium brine circulation. It is possible to obtain much lower temperatures by the use of chloride of calcium than common s[illegible] rine, −50° F. being the limit.

The following table gives the properties of chloride of calcium brine:

TABLE 29.

P[illegible] LCIUM BRINE.

Degrees Beaumé, 60° Fah.	Specific Gravity, 60° Fah.	Per Cent. of Calcium.	Fr[illegible]	[illegible]	Specific Gravity, 60° Fah.	Per Cent. of Calcium.	Freezing Point, Degrees Fah.	Specific Heat.
1	1.007	1	+31.[illegible]	[illegible]	1.169	19	+ 1.76	
2	1.015	2	30.38	[illegible]	1.179	20	− 1.48	
3	1.024	3	29.48	23	1.189	21	− 4.90	.76
4	1.032	4	28.58	24	1.199	22	− 8.68	
5.5	1.041	5	27.68	25	1.209	23	−11.64	.75
6.5	1.049	6	26.60	26	1.219	24	−17.14	
8	1.058	7	25.52	27	1.229	25	−21.82	
9	1.067	8	24.26	28	1.240	26	−27.04	
10	1.076	9	22.82	29	1.250	27	−32.62	
11	1.085	10	21.38	30	1.261	28	−39.28	
12	1.094	11	19.76	31	1.272	29	−46.30	.693
13	1.103	12	18.14	32	1.283	30	−54.40	
14.5	1.112	13	16.34	33	1.294	31	−52.42	
15.5	1.121	14	14.36	34	1.305	32	−39.28	
17	1.131	15	12.20	35	1.316	33	−25.24	
18	1.140	16	10.04	35.5	1.327	34	− 9.76	
19	1.150	17	7.52	36.5	1.338	35	+ 2.84	
20	1.159	18	4.64	37.5	1.349	36	+14.36	

The cost of chloride of calcium is about double that of salt. The quality is extremely variable; insist upon having *fused* chloride of calcium. The salt is excessively deliquescent, that is, it is capable of absorbing a large quantity of water; for this reason, it is often used as a drier. This great avidity of calcium chloride for water renders adulteration by the absorption of water very easy. Even the fused salt contains as much as 20 per cent. of water, whereas the unfused salt, though still in solid form, contains upwards of 50 per cent. of water. Care should therefore be used in selecting the salt.

When it is desired to purchase chloride of calcium, request samples. Dissolve a certain weight of each sample of the salt in the same quantity of water; take a hydrometer reading of each one of the samples after the salt is thoroughly dissolved; the one giving the highest reading is the best sample.

MAKING BRINE.

1438. When a plant is first charged with brine, the brine-pump delivery should be brought over the top of the brine tank, even though a temporary connection has to be made, and allowed to enter a crib. This crib should be about 2 ft. by 4 ft., and be hung from two stringers laid across the top of the tank. The bottom of the crib should be formed of slats and a piece of burlaps or bagging laid over them. The brine-pump is then slowly started, delivering a stream of brine into the crib. Into this crib, bags of salt are then dumped. In case chloride of calcium is used, the iron drums in which it comes should be cracked open by means of a sledge-hammer. It is best not to remove the iron until after the calcium is broken, the iron casing preventing pieces of the calcium from flying about. By keeping the pump running slowly, the salt in the crib will gradually dissolve. Samples of the brine should be drawn out occasionally and tested with a hydrometer until the required gravity is reached, when the addition of salt or chloride of calcium should be discontinued. The crib can then be removed and the temporary pump delivery disconnected. In case of chloride of calcium brine, all that is necessary to strengthen the brine is to throw some of the broken calcium into the brine tank, near the pump suction. The affinity of this salt for water is so great that it will readily dissolve, even when the brine is at a comparatively low temperature. Do not attempt to strengthen salt brine with chloride of calcium, expecting to gradually work a calcium brine into the system. The calcium will readily dissolve, but as it has a greater affinity for water than salt, it will precipitate the salt out of the brine, thus clogging the pipes in the circulation. If it is desired to make the change, pump out all the salt brine and mix up a fresh batch of calcium brine.

THE BRINE TANK.

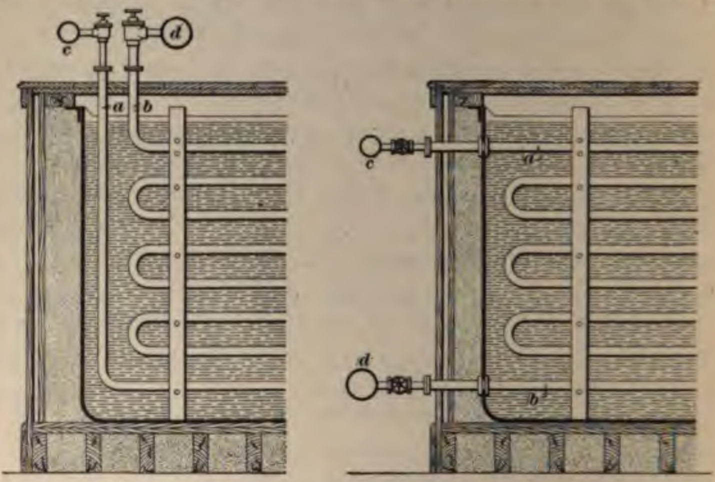
1439. The shape of the brine tank is preferably rectangular, and deep rather than wide. Sufficient headroom should be left above the tank for replacing the expansion coils. Depth is given to the tank so that the brine will have an opportunity of falling as it becomes colder, on account of its greater density. This arrangement permits the warm brine from the circulation to enter the tank near the top and the pump suction to take it out near the bottom, where the brine is the coldest. The tank should also be provided with a drain on the bottom for emptying it. The pump suction should be several inches above the bottom, so that any dirt, sediment, etc., may be able to settle on the bottom of the tank without entering the suction-pipe. It is also well to provide the suction with a strainer having a removable screen that can be drawn up through the top of the tank, cleaned, and replaced. Tanks 6 feet deep should be built of $\frac{3}{16}$-inch iron, riveted with $\frac{5}{8}$-inch rivets, $1\frac{1}{4}$ inches center to center. There should be a 3″ × 3″ angle-iron rim around the top, and one girth brace consisting of two 3″ × 3″ angle-irons with a $\frac{1}{2}$″ × 6″ plate riveted in between. This brace should be placed 30 inches from the bottom. The tank should be braced across the top every 6 feet. In case the tank is 8 feet deep, it should be constructed of ¼-inch iron, and it should be provided with two girth braces of 3″ × 3″ angle-iron with ½″ × 6″ plate. Brine tanks of the depths above specified should be provided with good foundations, as their weight when full exceeds the safe limit of the average warehouse floor. Preferably they should stand on brick or stone piers, carried up from hard-pan and covered with 2-inch planks. On these planks lay 3″ × 10″ joists, 12 inches center to center, well bridged, and fill the space between the joists with ground cork, or if that is too expensive, planing-mill shavings packed in tightly. Lay a ⅞-inch pine floor over the cork, then two layers of good water-proof paper, over which lay a 1¼-inch floor. The tank is then set on this floor and hot pitch is run in under it. The description of side-wall insulation is given in a subsequent section.

EXPANSION COILS.

1440. The shape of the brine tank governs the style of expansion coils to be used. In case of a round brine tank, it is necessary to use helical coils, one nested within the other. Such coils are unsatisfactory, as the inside coil is much shorter than the outside one, which makes it necessary to regulate the feed of each coil by itself, so as to properly distribute the ammonia between them. It is also difficult to clean these coils and repair them in case of leaks.

The simplest and most effective expansion coil is the flat return-bend coil. It is inexpensive to build, easily handled and shipped, and should be made of continuously welded pipe, with no joints in the tank. The tails of these coils should project through the side or top of the brine tank and should be fitted with expansion and stop valves and headers, as shown in Fig. 337.

The size of the main feed line supplying the flow header and that of the return or suction line from the return header should be the same as given in Art. **1435** for direct-expansion feeds and returns. The individual feed for each coil should be ½ inch. This is a convenient size, as it does not readily clog and is sufficiently small for fine adjustment; 1-inch pipe is a very satisfactory size to use for expansion coils, though larger sizes are often used.



Bottom Feed.
(a)

Top Feed.
(b)

FIG. 337.

To get economical results, coils of this size pipe should not exceed 150 feet in length. Fifty square feet per ton of refrigerating capacity and 90 square feet per ton of ice-making capacity should be allowed. In other words, one coil of 1-inch pipe 150 feet long will give 1 ton of refrigerating effect.

1441. There are three methods of supplying the expansion coils with anhydrous ammonia. They are:

1. The *bottom feed*, in which the liquid ammonia is admitted at the bottom of the coil and the suction is taken from the top. By this method, it is necessary to very nearly fill the coil with anhydrous ammonia, in order to get the benefit of all the pipe. Furthermore, the liberating area is equal to only the cross-sectional area of the pipe, so that the ammonia can not evaporate very rapidly.

2. The *top feed*, in which the liquid ammonia enters the top of the coil and the gas is taken from the bottom. This method does away with two of the important defects in the former method. A slight feed at the top of the coil allows a stream of liquid ammonia to gradually work down through the coil and evaporate along the full length of the coil. This gives a long evaporating area, and the benefit of the full length of the coil is obtained; further, the coldest ammonia is delivered at the top of the tank where it is most needed. The principal objection to this method is that there is a liability of overfeeding, which permits some of the liquid anhydrous ammonia to pass down into the return, and so to the compressor or absorber, without having evaporated.

The arrangement for the bottom-feed system is shown in Fig. 337 (*a*) and that for the top-feed system in Fig. 337 (*b*). In each figure *a* represents the feed-pipe, *b* the return pipe, *c* the flow header, and *d* the return header.

3. *Top feed* and *bottom expansion*, which is a combination of the two preceding methods, retains the good points of both. In this method, one coil is provided with a feed valve at the top, and the anhydrous ammonia is allowed to work through, as in case of the top-feed system; but instead of having the bottom tail of this coil connected with the return, this tail is connected with the bottom tail of the adjoining coil. Any liquid that is overfed in the first coil passes into the second, and there it is completely evaporated and passes out at the top into the return suction line. This last method, though more expensive than either of the others, gives very satisfactory results.

THE BRINE-PUMP.

1442. If chloride of sodium (common salt) brine is used, the brine-pump should be bronze-lined throughout; that is, it should have a brass cylinder, rods and glands on stuffing-boxes. This bronze lining is not necessary with chloride of calcium brine, iron working parts being perfectly admissible. Tupper's square flax packing is a good packing to use with chloride of calcium brine. Any ordinary soft packing will answer for salt brine.

The size of the brine-pump depends upon the quantity of brine circulated per minute and upon the range of temperature between the warm brine entering the expansion tank and the cold brine leaving it. A very good rule of thumb for capacity, in case of brine circulation, is as follows: *Twenty-five gallons of brine cooled one degree in one minute is equivalent to one ton of refrigerating effect per twenty-four hours;* hence 5 gallons of brine cooled five degrees or six gallons cooled four and one-sixth degrees are approximately equal to that tonnage.

If, therefore, it is desired to select a brine-pump for a 10-ton plant, and the range of temperature between the brine at the inlet and that at the outlet is 5°, a brine-pump of 50 gallons capacity should be selected. The maximum piston speed at which brine-pumps should run is 60 feet per minute; 40 feet per minute is better, and is a fair average. The pressure against which a brine-pump usually has to work is comparatively low, and the ordinary low service or tank pump is the type usually used. Both single and duplex pumps are used, the latter class predominating.

BRINE MAINS.

1443. In running brine mains, it is important to have the lines as straight as possible and to avoid risers and downtakes. Where these are unavoidable and form a loop, it is best to provide the highest point of the riser with an air-cock for bleeding it. This is particularly important in case of the secondary mains, where an air trap will cause the brine to flow through the other secondary mains at the expense of the one having the air trap. This gives rise to the supposition that the coils or main are blocked and that the circulation is prevented on that account. Water fittings of long radius or sweep are preferable for use in extended brine circulations; such fittings are considerably more expensive than the ordinary steam fittings, but are much better on account of the decreased friction. Unions used on brine work should in all cases be flanged unions or those having ground joints. A $\frac{1}{16}$-inch 2-ply rubber gasket is all that is necessary for the flange unions.

1444. In laying out a brine circulation, it is very important to distribute the flow of brine in such a way that the quantity sent to each portion of the house should return from all those parts at nearly the same return temperature. It is first necessary to estimate the tonnage required to cool each section; from this the quantity of brine is estimated. Knowing the quantity of brine, the next step is to ascertain the sizes of the various mains needed to conduct this quantity of brine with a friction head[*] of say 10 pounds per square inch. The method of calculating the tonnage will be given in a subsequent section under the different heads of cold-storage refrigeration, breweries, packing houses, etc The second step, namely, that of estimating the quantity of brine required, is based upon the rule of thumb given in Art. **1442.** The initial and terminal temperatures having been decided on, 25 divided by their difference gives the quantity of brine required per ton of refrigerating effect. This multiplied by the tonnage of any given section to be cooled gives the quantity of brine required to refrigerate that space.

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[*] When a liquid flows through a horizontal pipe with a uniform velocity, there is a loss of pressure due to the friction between the liquid and the pipe; thus, if the pressure of the liquid as it enters the pipe is 40 pounds per sq. in., it may be only 30 pounds on leaving the pipe, the loss being 10 pounds per sq. in. A pressure in pounds per square inch may be changed to a head of water as follows: A cubic foot of water weighs 62.42 pounds; therefore, a column of water one foot high with one square inch cross-section weighs $62.42 \div 144 = .43 +$ pound. It follows that a column of water produces at its base a pressure of .43 pound per sq. in. for each foot of height. The head of water corresponding to the loss of pressure when water or other liquid flows through a pipe is called the **friction head.** A friction head of say 20 feet corresponds to a loss of pressure of $20 \times .43 = 8.6$ pounds per sq. in.

This process may be expressed by the following formula:

$$G = \frac{25\,T}{t_2 - t_1}, \qquad (124.)$$

in which

G = gallons of brine required per minute;
T = tonnage of section to be cooled;
t_2 = temperature of brine inlet;
t_1 = temperature of brine outlet.

1445. The determ he size of the pipe necessary to carry a given nı rine is based on D'Arcy's formula for the flow oı wat ıgh clean cast-iron pipes. This formula is practic for wrought-iron pipes. In case there are very ma s in the work, it is best to deduct 10% from the resu ɔy the formula.

Let Q = gallons of bri ɛd per minute;
h = head in feet re overcome friction;
l = length of pipe

Then, for 1-inch pipe, $Q = 28.5\sqrt{\frac{h}{l}}$. (125.)

The formula as given applies only to 1-inch pipe. If it is desired to ascertain the quantity of brine that will flow through pipes of other diameters, it is necessary to multiply the amount that a 1-inch pipe will deliver by the factor in the following conversion table opposite the given diameter:

Diameter.	Factor.
1 inch	1.00
1¼ inches	1.84
1½ "	3.02
2 "	6.53
2½ "	10.23
3 "	19.10
4 "	40.50
5 "	72.00
6 "	115.00

If we denote by G the actual quantity of brine used per nute and by Q the quantity delivered per minute by a

1-inch pipe, then the factor required will be given by the quotient $\frac{G}{Q}$. The size of pipe required can then be determined from the value found for the factor.

EXAMPLE.—Two houses are supplied with refrigeration from a central station. The quantity supplied to house *A* is equivalent to the melting of 12 tons of ice, or, in ordinary language, *A* requires 12 tons of refrigeration; house *B* requires 14 tons. The range of temperature is 5°, the friction head from the station to *A* must not exceed 15 feet, and that from *A* to *B* must not exceed 12 feet. The distance of house *A* from the station is 150 feet and *B* is 200 feet beyond *A*. What size of pipe is required from the central station to *A*, and what size from *A* to *B*?

SOLUTION.—According to formula **124,** the quantity of brine that must pass per minute to house *A* is $G = \frac{25 \times 12}{5} = 60$ gallons, and for house *B*, $G = \frac{25 \times 14}{5} = 70$ gallons. Since the pipe to *A* must carry the brine for both *A* and *B*, it must carry 130 gallons per minute. To find the quantities delivered per minute by a 1-inch pipe under the given conditions, we use formula **125.** For house *A*, $Q = 28.5\sqrt{\frac{h}{l}} = 28.5\sqrt{\frac{15}{150}} = 9$ gallons, nearly. For *B*, $Q = 28.5\sqrt{\frac{12}{200}} = 7$ gallons, nearly. For house *A* the factor is $\frac{G}{Q} = \frac{130}{9} = 14.44$. The next higher factor given in the table is 19.1; hence a 3-inch pipe will be used. For *B*, $\frac{G}{Q} = \frac{70}{7} = 10$, which calls for a 2½-inch pipe between *A* and *B*.

BRINE COILS.

1446. The majority of brine coils are constructed of 1¼-inch pipe. The pipes are run in return-bend flat coils and hung on either walls or ceiling. The coils are built of common black steam or water pipe, and are provided with either cast or wrought iron return bends at the ends of the runs. Wrought-iron return bends are preferable, as there is less danger of leaks, due to sand holes, etc. Cast-iron couplings are used on well-built brine circulations, as pipe can be made up much tighter into these couplings without stretching them than in case of the wrought-iron couplings.

e return bends for 1½-inch pipe should be at least 6 inches ter to center, so as to give a good circulation of air between the pipes and prevent their freezing over into one solid mass. It will be seen that style (*a*), Fig. 338, gives a considerable more surface per lineal foot of pipe than style (*b*), where the runs are frozen together.

FIG. 338.

provided at inlet and outlet with he y steam-cocks. Gate-valves are not advisable, as the flow can be better regulated by a cock, and one can tell at a glance exactly how much the cock is open. Flange unions should be provided at both ends of the coil, just inside of the cocks, so that the coil can be readily disconnected in case of necessity. The inlet is usually connected at the bottom of the coil and the outlet at the top. A pet or air cock should be placed in the coil near the outlet cock for drawing off any air that may accumulate. In case side-wall coils are used, they should be hung on furring strips at least 6 inches deep, so as to keep the coil that distance from the wall. This will insure a good circulation of air and prevent the frost from reaching the wall. These furring strips should be 3″ × 6″ and be placed about 8 feet center to center. A drip pan or trough should be hung under the coil to catch the drip coming from the melted snow when the coil is turned off. It is best to have tubs at the ends of these troughs, as drain-pipes are liable to freeze up and cause trouble. Coils are usually run in lengths of about 300 feet of 1¼-inch pipe. Such a coil will absorb in twenty-four hours a quantity of heat equivalent to the melting of about ½ ton of ice. With a range of 4° between the inlet and outlet of the coil, about 3 gallons of brine a minute will be required to give this capacity. To secure this flow, the friction head need not exceed 2 feet.

1447. Testing.—When the brine circulation is com- ted, it should be subjected to a thorough test with hydraulic pressure equal to about double the running pressure of the brine. Leaks should be carefully repaired. When chloride of calcium brine is used, it is especially important to have a tight job of pipe fitting, as calcium brine never rusts up a leak. After the work has been completed, but before it has been accepted, the circulation should be subjected to a "frost test," as it is called. The brine is refrigerated as in regular service, and the brine pipes in the circulation including the mains are allowed to frost over. If there are any leaks, they will become apparent, as the leaky points will not frost and can be readily detected. It is best to leave off the insulation of the mains until a frost test has been applied, so that there is no chance for leakage and the consequent ruin of insulation.

COOLING THE BRINE.

1448. Ordinarily the brine is cooled in a brine tank containing expansion coils. This consists essentially of either a round or rectangular tank in which is placed one or more coils of pipe. The tank contains the brine solution, and the pipe coils the liquefied anhydrous ammonia, which expands in these coils and abstracts the heat from the brine. Two improvements on this system are known as the *Hendrick brine-cooler system* and the *British Linde circulating system.*

1449. Hendrick Brine Cooler.—The purpose of the brine cooler is to give a more efficient means of cooling brine than that of the ordinary brine tank. It consists of a cylindrical shell set on end and supplied with helical pipe coils continuously welded throughout their length. The tails of the coils pass through the top and bottom heads of the shell, and the ends of these tails are connected into a manifold or header. The shell is sufficiently strong to withstand a working pressure due to the back pressure of the machine. In effect the brine cooler is the reverse of the brine tank, the ammonia being on the outside and the brine inside of the coils. In this arrangement, the brine is pumped continuously through the coils, thus creating a very rapid circulation in them and giving a much higher efficiency to the surface than that in the brine tank, where the coils are submerged in a sluggish liquid, and the circulation depends upon the difference in the density of the brine as it becomes cold. Furthermore, the expansion of the ammonia in the shell insures the highest back pressure possible for any given temperature, and there is no chance of any throttling action in the coils, as in case of the expansion tank. This is a very important factor, especially in the case of compression machines, where the higher the back pressure the greater the efficiency of the machine.

Care is taken in designing coils for brine coolers of this type, so that the pressure required to overcome friction does not exceed ten pounds per square inch. Owing to the rapid circulation of both the ammonia, which boils around the coils, and the brine inside, it is possible to get an outlet brine temperature within 3° of the ammonia temperature. This insures a brine temperature nearly as low as the temperature of the ammonia itself, and brings up the efficiency of the brine system practically to that of the direct-expansion system.

The principal objection to the cooler is the friction in the coils, but this can be reduced to a minimum by proper design.

The circulation is similar to that of an ordinary brine circulation, but the connections are as follows: A brine-storage tank of any required capacity is used; the brine-pump takes the brine from this tank, and delivers it to the brine cooler. The cold brine leaves the bottom of the brine cooler and passes to the circulation coils, and the warm brine returns from the coils to the brine-storage tank.

1450. The British Linde System. — The *British Linde air-circulating system* consists of a series of flat return-bend direct-expansion coils. Brine is taken from a catch pan under these coils and is pumped over them. The air to be cooled is forced by means of a blower through these coils and leaves any moisture, together with any impurities, that it may contain in the brine. The arrangement is, in fact, the reverse of the atmospheric condenser, the air being cooled instead of heated, and brine taking the place of water. It is claimed for this system that it not only gives very dry air, but also that the brine absorbs any odors that air may be infected with. The air leaves the coils entirely free from any impurities—in fact so much so that eggs and lemons can be stored on the same system of air circulation. The principal objections to this system are the rapid corrosion of the pipes and the necessity of continually strengthening the brine charge. This is necessary, since the moisture that is abstracted from the atmosphere weakens the brine, and an addition of salt is necessary to keep it from freezing on the coils. The operation of this system is about as expensive as that of the ordinary brine-tank circulation.

TESTS OF REFRIGERATING-MACHINES.

GENERAL REMARKS.

1451. The primary object of a test of a refrigerating-machine is the determination of its efficiency—that is, the ratio of the refrigerating or ice-melting capacity to the fuel consumed. The test involves, therefore, the accurate determination of three principal data: (1) The capacity of the machine. (2) The heat given up to the condensing water. (3) The power consumed by the machine.

1452. An exact determination of the ice-melting capacity can not be made unless the machine is made to cool a liquid during the test. This restriction presents no difficulty in the case of machines working with the brine system, since the cooling of the brine is actually the work which the machine does. In direct-expansion machines, the capacity must be calculated from the weight and temperature of the ammonia circulated, though such a determination can not be considered accurate.

The heat given up to the condenser can readily be determined by weighing the condensing water and observing the temperatures at the inlet and outlet. For testing purposes, the condenser should be of the submerged type.

The work of compression may be determined by taking indicator-diagrams from the ammonia cylinder. Since, however, the work thus obtained would not include the work spent in overcoming the friction of the machine, it is not customary to base the consumption on the work of compression. In most compression machines, the engine and compressor are coupled together. The work shown by indicator-diagrams taken from the steam cylinder will include the work of compression and the friction both of compressor and engine. To determine the actual consumption of work by the refrigerating-machine, the work of the engine running empty may be determined by means of the indicator, and this subtracted from the total work of the steam cylinders will give the net work delivered to the machine.

In the case of absorption machines, the steam fed to the generator is measured, and furnishes a basis for comparison.

1453. Heat Balance.—The accuracy of the results of any test of a refrigerating-machine may be checked by forming a balance between the heat received and given up by the machine. It is clear that, on the whole, the heat received, including the heat equivalent of the work consumed, must be exactly equal to the heat rejected. In the case of the compression machine, the heat received from the brine or cold room is Q_2, the heat equivalent of the work of the compressor is $\frac{W}{J}\left(=\frac{W}{778}\right)$, and the heat given to the condenser is Q_1. These three quantities are measured directly, the work W in 24 hours being obtained from the indicator-diagrams taken from the compressor. Were it not for losses due to radiation, convection, etc., the observed quantities should satisfy the relation

$$Q_2 + \frac{W}{J} = Q_1.$$

If, however, we denote by Q_3 the total loss due to all causes, the heat balance is

$$Q_2 + \frac{W}{J} = Q_1 \pm Q_3.$$

In a carefully conducted test, the difference between the two members of this equation should be small. A large discrepancy indicates either some error in observation or gross inaccuracies in the apparatus or in the method of conducting the test.

TEST OF COMPRESSION MACHINE.

CAPACITY OF DIRECT-EXPANSION SYSTEM.

1454. As stated in Art. **1452,** an accurate determination of the quantity of refrigeration produced by a direct-expansion system is not possible, and it is necessary to resort to calculation to obtain an approximate result. From formula **123,** the refrigeration in 24 hours expressed in tons of ice is $F = .00505\ n\ M\ [r_1 - (t_c - t_a)]$, and the weight M of ammonia circulated per stroke is $M = \frac{C}{v} = C\ w$, where C denotes the volume of the compressor cylinder in cubic feet, v the volume of a pound of ammonia vapor at the pressure in expansion coil, and w the weight of a cubic foot of the vapor at the same pressure. Substituting, we have

$$F = .00505\ n\ C\ w\ [r_1 - (t_c - t_a)]. \qquad \textbf{(126.)}$$

The factor n is the number of compression strokes per minute; in single-acting compressors n will be equal to the number of revolutions per minute, and in double-acting machines to double that number. This formula gives the theoretical tonnage of the machine. To allow for such losses as clearance and cylinder superheating, deduct 25 per cent. from the result in case of a single-acting machine and 30 per cent. for a double-acting machine.

To obtain a rough approximation to the capacity, the following rule of thumb is sometimes used: With a suction pressure of 15 pounds, gauge, and a head pressure of 150 pounds, a well-made single-acting compressor with small clearance spaces gives an ice-melting effect of one ton for 4½ cubic feet of piston displacement per minute. With double-acting compressors, allow 5 to 6 cubic feet per minute for each ton.

EXAMPLE.—A single-acting compressor having two cylinders with 12-inch bore and 30-inch stroke is running at 60 revolutions per minute. The head pressure is 150 pounds and the back pressure 15 pounds. What capacity is the compressor developing? What would be the capacity of a single-cylinder double-acting compressor under the same conditions?

SOLUTION.—Referring to the table of the properties of ammonia, it is seen that t_a, the temperature in the expansion coil, is slightly below 0° F., and the temperature t_c, corresponding to the head pressure, is slightly below 85° F. It will be sufficiently exact to take these temperatures. The latent heat r, corresponding to 0°, is 555.5 B. T. U., and the weight of a cubic foot of ammonia vapor at 0° F. is .1109 pound. The volume of the compressor cylinder in cubic feet is $.7854 \times 1^2 \times 2\frac{1}{2} = 1.9635$. Substituting in formula **126**, the capacity for one cylinder is

$$F = .00505 \times 60 \times 1.9635 \times .1109 \times [555.5 - (85 - 0)] = 31.04.$$

For both cylinders the theoretical capacity is 62.08 tons, and deducting 25 per cent., the approximate actual capacity is 62.08 tons × .75 = 46.56 tons. Ans.

Evidently the theoretical capacity of the double-acting single-cylinder compressor would also be 62.08 tons; the probable actual capacity, deducting 30 per cent., is 62.08 tons × .70 = 43.46 tons. Ans.

1455. Direct Measurement of Ammonia.—The formula for capacity applies to compressors working under the most favorable conditions, with small clearance losses and no leak about valves or pistons. The cylinder superheating is based upon a suction pressure of 15 pounds and a back pressure of 150 pounds. This loss is found in case of wet or dry compression or oil injection, the exact percentage differing with make of machine. To make even an approximately exact test upon a compression machine running with a direct-expansion system, it is necessary to measure directly the quantity of ammonia circulated by means of an ammonia meter in the feed line, between the receiver of the condenser and the expansion-valve.

The meter should be calibrated in cubic feet by means of water before being placed in the feed line, care being taken to thoroughly eliminate any remaining water before the meter is connected up. The weight in pounds per cubic foot of the liquid anhydrous ammonia for any required head pressure is given in the table of Properties of Saturated Ammonia.

Let P denote the weight in pounds of anhydrous ammonia passed through the meter in one hour. Then the heat absorbed by the ammonia per hour is $P\,[r - (t_c - t_a)]$ B. T. U. and the capacity in tons per 24 hours is

$$F = \frac{24}{285,300} P[r - (t_c - t_a)] = .000084\, P[r - (t_c - t_a)]. \qquad \textbf{(127.)}$$

EXAMPLE.—If an ammonia meter shows a delivery of 25 cubic feet of liquid anhydrous ammonia per hour, the machine running with a back pressure of 15 pounds and a head pressure of 150 pounds, what is the actual capacity of the machine?

SOLUTION.—Taking the temperatures t_a and t_c at 0° and 85°, as in the solution of the example of Art. **1454,** the volume of a pound of liquid at the given head pressure is .0273 cubic foot; therefore, 25 cubic feet will weigh $\frac{25}{.0273}$ lb. = 916 lb. The latent heat r is 555.5 B. T. U.; hence, substituting in formula **127,**

$$F = .000084 \times 916 \times [555.5 - (85 - 0)] = 36.2 \text{ tons.}$$

CAPACITY OF BRINE SYSTEM.

1456. When the refrigerating-machine is used to cool a liquid—as, for example, brine—the quantity of refrigeration produced can be determined by direct observation. If we denote by G the weight of brine circulated in a given time, by s the specific heat of the brine, and by t_1 and t_2, respectively, the temperatures of the brine at the outlet and inlet, then, according to formula **72,** *Heat*, the heat given up by the brine in the given time is

$$G\,s\,(t_2 - t_1) \text{ B. T. U.}$$

The heat given up in 24 hours is readily calculated, and from this the capacity in tons follows at once.

ILLUSTRATION.—In one of Prof. Denton's tests previously mentioned, the average temperature at the brine inlet was 36.76° F., that at the outlet was 28.86° F., the specific heat of the brine was .82, and the weight of brine circulated per minute was 2,281 pounds. The heat given by the brine to the ammonia per minute was, therefore,

$$2{,}281 \times .82 \times (36.76 - 28.86) = 14{,}776 \text{ B. T. U.}$$

1457. Arrangement for Test.—Fig. 339 shows the general arrangement for making a capacity test on machines of any type that use brine as a carrier of cold.

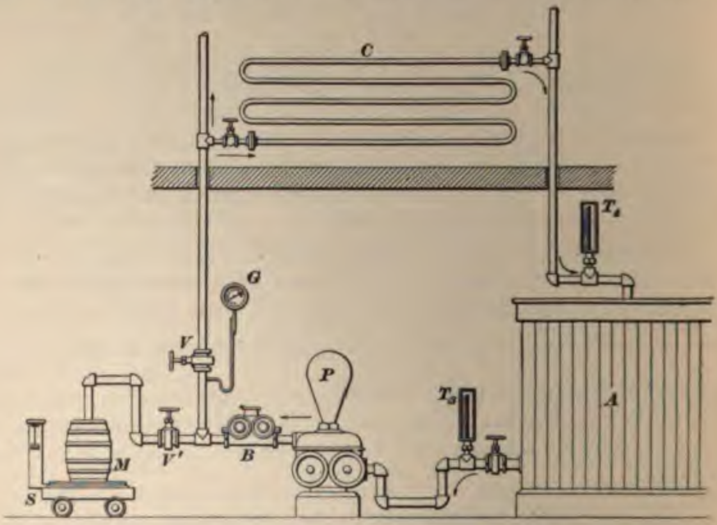


FIG. 339.

The brine tank is shown at *A*, the brine-pump at *P*, the brine meter at *B*, and at *C* the coils in the cooling rooms. A tee should be placed in the delivery line between the brine meter and coils *C*, and cocks or valves should be placed on each side of this tee. These cocks are denoted by *V* and *V'*. A pressure gauge *G* indicates the pressure of the brine. A tank or large barrel *M* is placed on the scale *S*, and is so arranged that the pipe from *V'* leads to it. A thermometer T_4 is placed in the brine line returning to the tank, and another T_3 is placed in the pump-suction line leaving the tank.

1458. Thermometers and Mercury Wells. — Thermometers used in making tests on refrigerating work should be very accurate; they should be graduated to tenths of a degree, so that they can be read to hundredths of a degree. Iron sockets, as shown in Fig. 340, having a ¾-inch pipe thread near the top, should be placed in the pipe wherever temperatures are required. The socket is filled with mercury and the bulb of the thermometer is placed in it. The iron around the socket should be thin, so that the heat may be readily transmitted. The socket should be at least 3 inches long, and if the pipe is not of sufficient diameter, larger size tees with outlets bushed should be provided to receive them. Where it is desirable to get an approximate temperature, portable sockets or wells made of wood

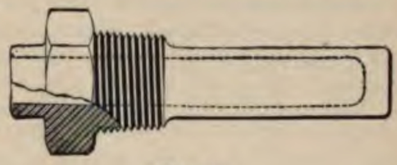


FIG. 340.

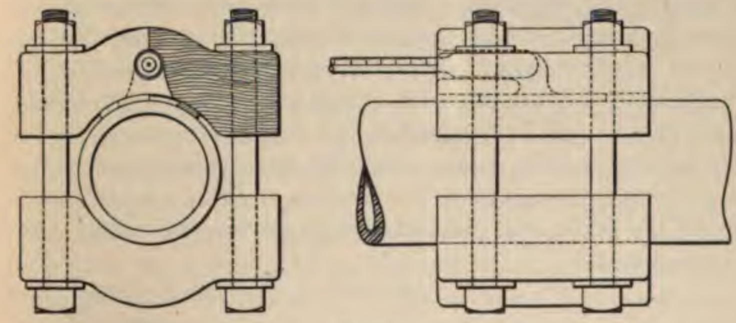


FIG. 341.

can be used with very good results. Fig. 341 shows a sketch of such a socket for vertical pipes.

1459. Calibrating the Brine Meter.—The specific gravity of the brine should be determined by a specific-gravity hydrometer at 62° Fahrenheit. The corresponding specific heat for brine at this gravity can be found by referring to Table 28 or 29, according to the kind of brine used. There now remains to be determined the number of pounds of brine that the meter discharges per cubic foot. To do this, the brine is circulated through and the reading of the pressure gauge *G*, Fig. 339, is noted. The line from the valve *V'* to the barrel should be arranged with a swinging joint, so that it can be swung over the barrel or beyond it at will. Stop the brine-pump, close the valve *V*, and swing the pipe beyond the barrel. Start the brine-pump and open the valve *V'* until the gauge *G* shows the same pressure as in regular running. Throw the pipe into the barrel, and at the same instant take the reading of the meter *B*. When the barrel is nearly full and the reading of the meter has come to an even number of cubic feet, throw the pipe clear of the barrel and weigh the amount of brine held by the barrel. The barrel should be weighed empty before the test is started. The difference between these two weighings will give the weight of the number of cubic feet of brine shown by the meter reading. For example, if the net weight of the brine is 736 pounds and the meter reading is 10 cubic feet, 73.6 pounds is the weight of 1 cubic foot according to the meter. Such a determination does away with the necessity of multiplying the meter reading by the specific gravity of the brine, and also detects any error that may exist in the meter.

DETERMINATION OF HEAT REJECTED.

1460. The procedure adopted in measuring the heat rejected by the ammonia to the condenser is exactly similar to that employed in measuring the heat absorbed from the brine. The quantity of water passing through the condenser is indicated by a water-meter, previously calibrated. The temperatures of the water at the entrance and at exit are measured by accurate thermometers. Evidently the product of the weight of water passing through the meter in a given time and the range of temperature give the heat absorbed in that time in B. T. U. If the compressor has a water-jacket, the heat delivered to the water of the jacket is determined in a similar manner.

ILLUSTRATION.—In Denton's test, quoted in a preceding paragraph, the average initial temperature of the condensing water was 44.65° F., the average final temperature was 83.66° F., and the quantity circulating through the condenser per minute was 2,281 pounds. The average temperature of the water entering the jackets was 44.65°, that of the water leaving the jackets was 69°, and the weight passing through the jackets per minute was 25 pounds. The heat rejected to the condenser per minute was

$$2,281 \times (83.66 - 44.65) = 88,981.8 \text{ B. T. U.}$$

The heat rejected to the water in the jackets per minute was

$$25 \times (69 - 44.65) = 609 \text{ B. T. U.}$$

MEASUREMENT OF THE WORK OF COMPRESSION.

1461. The work of compression and the power consumed by the compressor are determined from indicator-diagrams taken from the compressor cylinder. All the directions for taking diagrams given in *Steam and Steam Engines*, Arts. **1259-1261**, apply equally well to refrigerating-machines.

For ammonia work, it is preferable to use a special indicator, the working parts of which are made of steel instead of brass. Ammonia has no effect on steel, but rapidly corrodes brass. In case it is not possible to procure an ammonia indicator, an ordinary steam indicator will answer the purpose, provided the piston is removed after every series of cards is taken and both cylinder and piston are wiped dry and well covered with oil. This will prevent the ammonia gas from attacking the portions of the indicator made of brass.

In the case of a vertical, single-acting compressor, the indicator pipe is screwed into the upper head of the cylinder.

If the compressor is double-acting, it is of course preferable to use two indicators, one at each end of the cylinder; but if only one indicator is at hand, the pipes from the two ends may be joined by a three-way cock, as shown in Art. **1259**, *Steam and Steam Engines*.

If the condensing pressure is below 150 pounds, a 100-pound spring will do the work; if above that pressure, a 125-pound spring should be used.

The diagrams may be worked up as explained in *Steam and Steam Engines*, Arts. **1265** to **1268.** Having obtained the mean effective pressure, P, the indicated horse-power, is given by the formula

$$\text{I. H. P.} = \frac{P L A N}{33,000};$$

the work per minute in foot-pounds is expressed by the product $P L A N$; the work in 24 hours in foot-pounds is

$$W = 24 \times 60 \times P L A N = 1,440\, P L A N,$$

and the heat equivalent of this work is

$$\frac{W}{J} = \frac{W}{778} = \frac{1,440\, P L A N}{778} \text{ B. T. U.}$$

The factor N in the preceding formulas denotes the number of *compression* strokes per minute; in the case of single-acting compressors, N is equal to the number of revolutions per minute, and in the case of the double-acting compressors, it is double that number.

1462. Examples of Indicator-Diagrams. — The diagrams shown in Figs. 326 and 327 are excellent examples of indicator-diagrams for a compressor having very small clearance spaces. It will be noticed that, though there is some clearance space, the heel of the diagram is very sharp, there being no curve similar to the compression curve on a steam-engine diagram. This is due to the fact that the ammonia gas in the clearance space is cooled sufficiently by the water-jacket to drop to the suction pressure by its own contraction, due to the loss in temperature, before the piston has receded any appreciable amount. The pressure at the end of compression shows that the compressor is obliged to compress the gas a little above the condenser pressure before the valve opens; the same is true in the case of the suction-valve, as the heel of the card falls below the suction pressure for a short distance until the suction-valve opens.

SCHEDULE OF TEST.

1463. In making a complete efficiency test of a compression refrigerating-machine, the data and results contained in the items of the following schedule should be obtained. In making a capacity test only, the items marked * need not be observed.

GENERAL DATA.

1. Date.
2. Duration of test.
3. Name of machine.
4. Class of machine.
5. Nominal capacity of machine.
6. Diameter of steam cylinder.
7. Stroke of steam cylinder.
8. Diameter of ammonia cylinder.
9. Stroke of ammonia cylinder.
10. Diameter of brine-pump, steam end.
11. Diameter of brine-pump, brine end.
12. Stroke of brine-pump.

OBSERVATIONS.

13. Average high ammonia pressure, gauge.
14. Average back ammonia pressure, gauge.
15. Average temperature of the brine inlet.
16. Average temperature of the brine outlet.
17. Average range of temperature of brine.
18. Weight of brine circulated per minute.
*19. Average temperature of condensing water at inlet.
*20. Average temperature of condensing water at outlet.

*21. Average range of temperature of condensing water.

*22. Weight of water circulated per minute through condenser.

*23. Weight of water circulated per minute through jackets.

*24. Average temperature of water entering jackets.

*25. Average temperature of water leaving jackets.

*26. Average range of temperature in jackets.

27. Average temperature in engine room.

28. Specific gravity of brine.

29. Specific heat of brine.

30. Revolutions per minute.

*31. Mean effective pressure, steam cylinder.

*32. Mean effective pressure, ammonia cylinder.

Results.

*33. Average horsepower of steam cylinder.

*34. Average horsepower of ammonia cylinder.

*35. Average friction horsepower.

*36. Friction horsepower in per cent. of steam horsepower.

*37. Condensing water, gallons per minute per ton.

38. Ice-melting capacity, tons per twenty-four hours.

*39. Refrigerating effect, pounds of ice, per pound of coal.

Heat Balance.

40. Heat given to ammonia by brine per minute, B. T. U.

41. Heat given to ammonia by compressor per minute, B. T. U.

42. Total heat received by ammonia per minute, B.T.U.

43. Heat delivered to condenser by ammonia per minute, B. T. U.

44. Heat delivered to jackets by ammonia per minute, B. T. U.

45. Total heat rejected by ammonia per minute, B.T.U.

46. Difference between heat received and rejected, B. T. U.

CONDUCTING AND WORKING UP THE TEST.

1464. Before a test is started, the plant must be properly arranged. Meters must be standardized, thermometer sockets must be placed, and accurate low-temperature thermometers must be obtained and compared with a standard. For accurate determination of the capacity, the test should not be shorter than ten hours, while official tests are seldom shorter than twenty-four hours, and are often continued for several days.

The frequency of the observations depends somewhat upon the length of the test and upon the uniformity of the conditions. Brine temperatures, items 15 and 16, should be read every 15 minutes; all temperatures should be read to tenths of a degree, and closer if possible. The necessity of this will be easily seen when one considers that the usual range of temperature is four or five degrees Fahrenheit, and the discrepancy of a tenth of a degree would mean from two to three per cent. variation in the results.

Readings of pressure gauges and meters may be made every half hour. The inital and final readings of the meters should be made with particular care. It is best to have two persons make these readings where possible. Indicator-diagrams should be taken every half hour or hour.

The specific gravity of the brine, item 28, is determined by means of an accurate hydrometer; the specific heat, item 29, may be found from Table 28 or Table 29, according as the salt or calcium chloride brine is used.

After the test is completed and the observations are recorded, the *average* of all the observations on each item should be found, and the items included under "Results" should be calculated. Items 33 and 34 are obtained by measuring the indicator-diagrams; item 35 is the difference between items 33 and 34, and item 36 is obtained by dividing item 35 by item 33. To find item 38, we have the equation

$$\text{Item } 38 = \frac{\text{item } 18 \times \text{item } 17 \times \text{item } 29 \times 60 \times 24}{285{,}300}.$$

(See Art. **1456.**)

Item 37 is readily obtained from items 22 and 38.

Items 40 to 46 can readily be calculated from the principles already explained. Should there be a considerable discrepancy between items 42 and 45, that is, should item 46 be relatively large, this fact would indicate either inaccurate apparatus or errors in the observations.

CALCULATION OF EFFICIENCY.

1465. In order to determine the commercial efficiency of a refrigerating plant, it is necessary to make a capacity determination, and at the same time take into account the amount of fuel consumed to produce the work. The ratio between these quantities will give the efficiency of the *plant*, as explained in Art. **1342**; but it will not determine the efficiency of the *machine*, as it takes into account all the steam consumed by the auxiliaries, such as the water-pump, brine-pump, boiler-feed pump, etc. To determine the efficiency of the machine, including the steam engine, only one of the following plans may be adopted:

1466. Approximate Commercial Efficiency.—The average horsepower of the steam cylinder is determined as indicated in item 33, and the fuel consumption per horsepower is judged from the type of the engine and the conditions under which it is working. The consumption of engines ordinarily used to drive refrigerating-machines varies from 2½ to 4 pounds of coal per horsepower per hour, the lower figure applying to condensing engines. It is quite customary to assume a consumption of 3 pounds per horsepower. Using this figure, we have the following formula:

$$\text{Efficiency (item 39)} = \frac{\text{item } 18 \times \text{item } 17 \times \text{item } 29 \times 60}{\text{item } 33 \times 3 \times 142.65}.$$

EXAMPLE.—In an actual test, the following data were recorded:

Brine circulated per minute	1,030 pounds.
Average range of brine temperature	8.62°.
Specific heat of brine	.78.
Average horsepower of steam cylinder	61.79.

Assuming a fuel consumption of 3 pounds of coal per horsepower per hour, find the efficiency of the combined refrigerating-machine and steam engine, expressed in pounds of ice melted per pound of coal.

SOLUTION.—

$$\text{Heat abstracted per hour} = 1{,}030 \times 8.62 \times .78 \times 60 \text{ B. T. U.}$$

$$\text{Ice melted per hour} = \frac{1{,}030 \times 8.62 \times .78 \times 60}{142.65} \text{ pounds.}$$

$$\text{Coal used per hour} = 61.79 \times 3 \text{ pounds.}$$

$$\text{Ice melted per pound of coal} = \frac{1{,}030 \times 8.62 \times .78 \times 60}{61.79 \times 3 \times 142.65} = 15.71 \text{ pounds.}$$

Ans.

1467. Direct Measurement of Steam and Fuel.—The result obtained by assuming the coal consumption for the engine can only be approximate, and a comparison between two machines based upon such an assumption might easily be unfair to one or the other of them.

In order to make an accurate efficiency test on any particular plant, the exhaust of the compressor engine should be condensed and weighed. To do this, an arrangement similar to the one shown in Fig. 342 will be found convenient. *A* is a steam condenser of any convenient form, and *B* and *B'* are barrels placed upon the scales *S* and *S'*. The outlet from the condenser *A* is provided with a pipe having a swinging joint *D*, so that it can be readily shifted from one barrel to the other. Before the test, the barrels

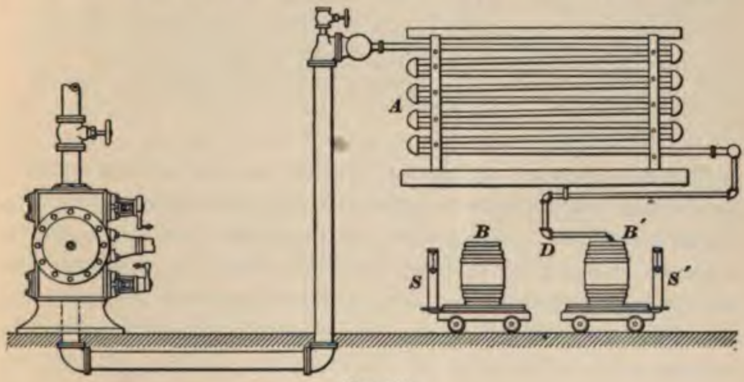


FIG. 342.

should be filled with warm water, so that they will absorb all moisture possible, and then should be weighed empty. When the test is started, the water is allowed to enter the barrel *B* until that barrel is full. The pipe *D* is then quickly swung to barrel *B'*, and while that is filling, barrel *B* is weighed. The barrels are thus alternately filled, weighed, and emptied, until the end of the test. The total amount of water condensed and weighed in the barrels is calculated, and if the test is for a shorter time than twenty-four hours, the proportionate amount for the full twenty-four hours is estimated. The weight of water used in twenty-four hours divided by the capacity in tons gives the weight of steam consumed per ton of refrigerating effect per twenty-four hours. If a boiler test is carried on at the same time that the refrigerating-machine is being tested, the evaporation of the boilers can be determined. The total amount of steam used in twenty-four hours divided by the weight of coal required to evaporate a pound of water will give the number of pounds of coal required to operate the refrigerating-machine for that space of time. The efficiency in pounds of ice per pound of coal can then readily be calculated.

1468. Power Plants.—Plants run by belt or water-power can best be tested by indicating the ammonia cylinder and estimating from the resulting cards the horsepower required to operate the plant. From this, the capacity per horsepower expended can be easily estimated. An allowance of at least 10 per cent. may be made for the friction of the machine.

TEST OF ABSORPTION MACHINE.

1469. As there are very few absorption machines on the market that make perfectly anhydrous ammonia, the only satisfactory way to make a test on machines of this class is by means of brine circulation. The arrangement of the connections will be the same as for the compression machine, and the data required and the method of calculation are the same for both systems. In case an absorption machine is used in connection with direct expansion, and it is not possible to make a thermal-heat test, one of the following methods may be used for the purpose of establishing approximate results.

1470. Measuring the Anhydrous Ammonia.—The anhydrous ammonia may be measured by a meter in a manner similar to that described in Art. **1455,** but samples of this ammonia should be frequently drawn off and tested by allowing the sample to evaporate in a flask, as described in Art. **1364.** If any moisture is found, the test will be of little value, as such a small quantity as 5 per cent. of entrained moisture will cut down the efficiency of the machine 20 per cent. and the capacity 10 per cent.

1471. Measuring the Pump Delivery.—The other method consists in measuring the pump delivery. If, however, it is not convenient to provide an ammonia meter, the ammonia-pump strokes can be counted, and the amount of ammonia actually discharged can thus be estimated.

Before starting this test, see that the packing-rings in the ammonia end of the pump are tight, so that there is no leakage from one side of the pump to the other; also that all the valves are seated and tight on their seats. The exact stroke of the pump should then be taken every few minutes, while the pump is in operation, so as to see if there is any variation, which is often the case with steam-pumps. After the exact length of stroke has been determined, the cubical contents of the pump can readily be computed. A revolution counter should then be attached to the piston rod so as to record the number of displacements made by the pump during the test.

1472. A general arrangement of the plant is shown in Fig. 343, in which *A* is the absorber of the machine to be tested and *B* is the strong-liquor receiver connected with the ammonia pump *P* by the pipe *S*. A tee is provided in this pipe, the opening of which is controlled by the valve *C*. The outlet of this valve is connected with a small coil *D* immersed in a pail of ice-water *E*. If the pump suction *S* is very warm, a sample drawn out of *E* will be cooled in passing through the coil *D*. It is necessary to cool this sample so that the strong liquor in the absorber under several pounds pressure will not lose any of the gas when reduced to atmospheric pressure. If the liquor is chilled some 15° or 20°, it will retain this gas, whereas if the sample be drawn out at the temperature of the absorber, the gas will leave the liquid as it is being drawn out, and the sample thus obtained will be much weaker than the liquor in the absorber. The same is true with the weak-liquor line *W*, which is also arranged with a pipe coil having an outlet *H*. In drawing samples from either of these two sources, care should be taken that all the liquor left in the coil should first be run out before a new sample is taken.

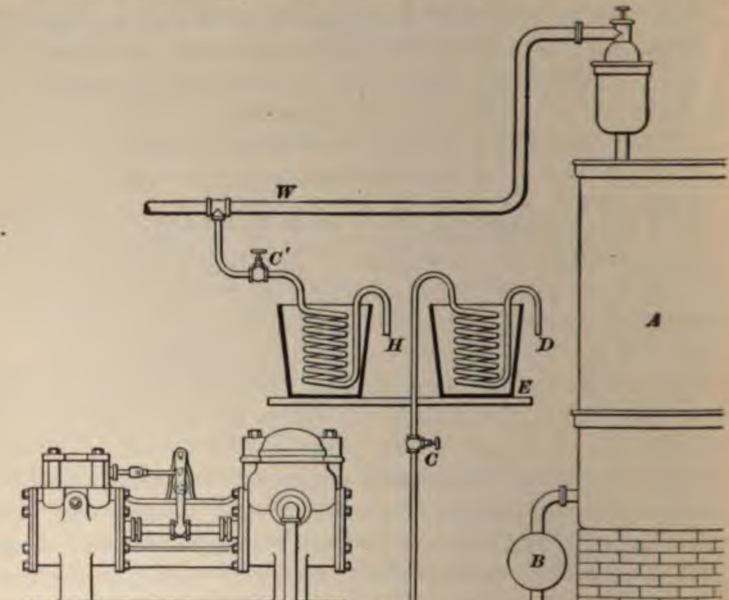


FIG. 343.

1473. In making hydrometer readings, the following points should be observed: In filling the hydrometer jar, let the inlet pipe *D* or *H* enter the jar so that it nearly touches the bottom. This will prevent any foaming or escape of gas from that cause. When the jar is full, let a little of the liquid run over the edge for the purpose of removing any foam, air-bubbles, etc., from its surface. The hydrometer is then carefully lowered and a reading is taken. Depress the hydrometer half an inch or an inch after making the first reading, and then allow it to rise. Then make another reading, and the mean of these two readings should be the correct reading of the liquid. The object of the double reading is to eliminate the effect of capillary attraction.

The temperature of the liquid should be taken at the time of the hydrometer reading and a correction should be made as follows: If the temperature of the liquid is above 55° F., 1 degree Beaumé should be deducted for every 17° above, and if the liquid is colder than 55°, 1 degree Beaumé should be added for every 17° below 55°.

If the machine is running with a comparatively steady load, samples of liquid should be taken from the strong and weak liquor lines every 15 minutes; their specific gravities should be determined, and at the end of the test the average gravity of each of the two liquors should be computed. Table 27 should then be consulted to determine the per cent. of ammonia in both the strong and the weak liquor. The difference between these readings will give the per cent. by weight of ammonia in each pound of strong liquor pumped. This per cent. of ammonia multiplied by the total weight of strong liquor pumped in an hour will give the weight of anhydrous ammonia circulated through the system in that time. Formula **127** can then be used for determining the capacity. In making the test, it is assumed that the machine is making anhydrous ammonia and that there is no entrained moisture.

1474. Economy of the Absorption Machine.—Having measured the capacity of the absorption machine, preferably by brine circulation, the efficiency or economy of the system remains to be determined. We have in this case two sources of steam consumption, namely, the generator and the ammonia-circulating pump. The consumption of the latter, in case it is a steam-pump, can be determined by an arrangement similar to that used in connection with the steam engine of the compression machine, and shown in Fig. 342. If, however, the pump is driven by an engine that also does other work, the best way to determine its steam consumption is as follows: Indicate the engine with the pump running when it is doing normal work, and indicate it again under the same conditions, but with the ammonia pump disconnected. Compute the horsepower from each set of diagrams; the difference between the two will give the power necessary to drive the ammonia pump. From this the steam consumption can be estimated more or less closely from the class of engine which drives the pump.

To ascertain the amount of steam taken by the generator, the arrangement shown in Fig. 344 may be adopted. The

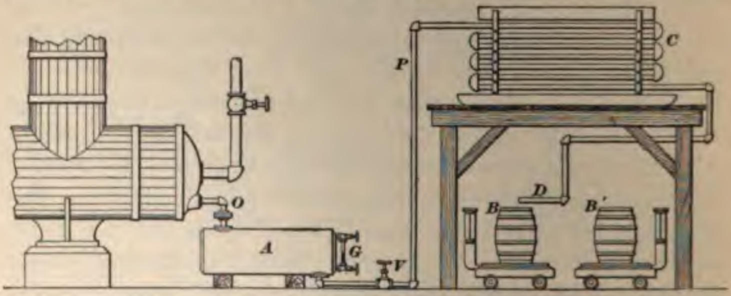


FIG. 344.

generator coils are connected by a pipe *O* to the receiver *A*, which is supplied with the gauge-glass *G*. The bottom of *A* is connected with a cooling coil *C* by a pipe *P*, in which is placed a valve *V*. The outlet *D* of this coil is arranged to swing over the barrels *B* and *B'*, which are placed on scales. The condensed steam entering *A* can be seen in the glass *G*. This is kept at a constant level by means of the valve *V*, so that only the amount of steam used in the generator will pass into the coil C, and from there into the barrels B and B'. The barrels are then manipulated as explained in Art. **1459** in connection with the compression machine. The weight of steam consumed being known, the calculation for efficiency proceeds as explained in Art. **1467.**

1475. Heat Balance. — The various quantities of heat with which we are concerned in the test of an absorption machine are the following: (1) The heat given up to the condensing water by the condenser, absorber, rectifier, and weak-liquor cooler; let this heat be denoted by Q_c. (2) The heat absorbed in the generator, Q_g. (3) The heat abstracted from the brine, Q_b. (4) The heat equivalent of the work of the pump, which may be denoted by Q_p.

To determine the quantity Q_c it is necessary to measure the weight of condensing water flowing per minute or per hour and the temperature range. (Items 17 and 18, Schedule of Test.) The heat Q_g may be determined as follows:

Let S = weight of steam passing through the condenser per minute;
L = latent heat of steam at given pressure;
t_1 = temperature of steam at given pressure;
t_2 = temperature of water leaving generator.

In changing to water, each pound of steam gives up L B.T.U. and the water gives up in addition $t_1 - t_2$ B.T.U. in passing from the temperature t_1 to t_2 (assuming, as is customary, that the specific heat of water is 1).

Therefore,

$$Q_g = S\,(L + t_1 - t_2). \qquad \textbf{(128.)}$$

The heat Q_b is determined as in the test of the compression machine, and the heat Q_p is found from the pump indicator-diagrams.

The total heat given to the machine is evidently

$$Q_g + Q_b + Q_p,$$

and Q_c is the heat delivered by the machine to the condensing water; hence we have

$$Q_c = Q_g + Q_b + Q_p.$$

In practice, Q_p is so small that it may be neglected in forming the heat balance, and the equation becomes

$$Q_c = Q_g + Q_b.$$

REFRIGERATING
AND
ICE-MAKING MACHINERY.

INSTALLATION AND MANAGEMENT OF REFRIGERATING-MACHINES.

COMPRESSION MACHINE.

ERECTION.

1476. Foundations.—The foundation for the compression machine consists of a regular engine foundation for the compressor, engine, and pump, and the necessary floor for the condenser, on which is placed the catch pan, if the condenser is of the atmospheric type. In the case of a submerged condenser, a stone foundation or brick piers should be provided. For brine circulation, the brine tank should also be set upon a strong brick or stone foundation. The footing course of the engine or compressor foundation should be of large stones or of a grout foundation well tamped. The upper courses of the foundation may be of either concrete or stone, the latter being preferable. In setting the foundation bolts, it is well to set them in 1½ or 2 inch wrought-iron pipe, which gives a limited play to the foundation bolt and allows it to enter the base of the engine or compressor more readily, particularly if the template is not true. Care should be taken to wedge the head of the bolt that remains in the foundation, to prevent its turning.

After the masonwork is carried to the proper height, a 1½ or 2 inch hardwood plank should be placed between the stone foundation and the bed-plate of the machine. This plank acts as a cushion and greatly diminishes the injurious effects of shocks or pounding. This end would not be attained if the machine were set directly on the stone foundation, owing to the lack of elasticity of the latter.

1477. Lining Up and Cleaning.—In case of a self-contained machine, where the engine and compressor are tandem and on the same bed-plate, there is no need of lining up, as these parts are usually shipped from the shop in perfect alinement. But in case of the cross machine, with the engine parallel with the compressor, great care should be taken in lining up both the compressor and the engine with the main shaft and in making the foundation of equal strength throughout, so as to prevent any possibility of either part settling more than the other. A slight settling of the engine or compressor will throw it completely out of line with the other parts, and this causes a great deal of trouble and annoyance, to say nothing of excessive oil bills.

After the engine and compressor are practically in line, all bearings should be thoroughly cleaned with kerosene to remove all grit and then keyed up to a loose fit. It is better to have the compressor knock a little at the start rather than run any risk of burning out the brasses or melting the babbit. The valve bonnets and valves should be removed and the cylinder thoroughly inspected. The castings in the suction and discharge ports should be kept perfectly clean, as the smallest particle of any substance lodging under the valves naturally causes a leak and is likely to cut either the valve, the cylinder, or both.

1478. Placing the Condenser.—After the engine and compressor are in place and bolted down and all bearings protected, so as to prevent dirt entering, them the next thing is to put the condenser in place. Whether the latter is of the submerged or of the atmospheric type, it is essential that the foundation or floor on which it is to stand should be practically level. Any inequality in the floor causes a like unevenness in the pipes or coils of the condenser. This naturally makes a trap in which the anhydrous liquid can collect, thereby interrupting the work of the condenser. The floors of an atmospheric condenser are often made to pitch in a certain direction, at right angles to the run of the coils. In such cases, the coils are practically level, but each coil is propped up a certain amount so as to allow for the pitch in the floor. With submerged condensers, it is best to have very deep tanks, as the deeper the tank the greater is the range in the temperature of the water. Such condensers always have the water-supply near the bottom, the overflow or waste being at the top. In this case, it is necessary, as already stated, to have very strong foundations for the condenser tank. These should be brick piers placed on good stone or concrete bases, and sufficiently close on centers to prevent any sagging in the floor beams that support the condenser tank. The coils in the tank should also be carefully leveled, so as to prevent any traps or pockets in the pipes, and the best practice recommends valves or cocks at the ends of each coil, outside of the tank, as in this way any leaking coil may be shut off independently of the others.

1479. Setting the Brine Tank.—As the brine tank is usually made deep and has considerable insulation, it is advisable to set it on a brick or stone foundation, similar to the one described in the preceding article. On top of this foundation, the insulation for the bottom of the brine tank is built in the form of a floor. The best thing to use for the filling between the joists is granulated cork; but this is very expensive, and planing-mill shavings are often used instead. Care should be taken not to use sawdust, owing to the possibility of spontaneous combustion in case it becomes dampened, and also because of its great avidity for moisture.

A well-insulated brine tank is shown in Fig. 345. It consists of the main joists resting on the foundations, two air spaces, and one-half to 1 inch of pitch, in which the brine tank is bedded. In case it is not possible to procure pitch, the floor on which the brine tank is to rest may be sprinkled with either ground cork or planing-mill shavings about 1 inch thick, and the brine-tank bed set in them. When the brine tank is rectangular and is to be placed in a corner, the walls round it should be insulated before putting the tank in place.

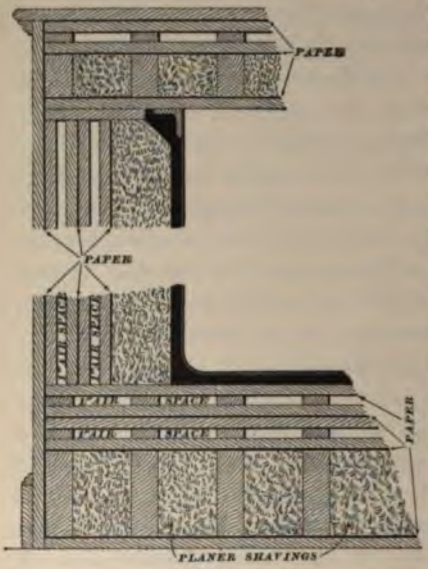


FIG. 345.

The tank should be thoroughly tested for leaks before it is lowered on the foundation, and if any leaks are discovered, they should be stopped.

After the tank is lowered, the expansion coils are placed in it, the same care being taken to keep them level as in the case of the condenser coils. Sometimes cheap plants are built without cocks or valves in the expansion coils; but the better class of machine builders put expansion and shut-off valves in each separate coil. This facilitates regulation and saves time and money in case of accident, as each coil can be shut off independently.

The insulated cover of the brine tank need not be as heavily insulated as the bottom and sides. It should be provided with a manhole or trap-door to give access to the inside of the tank. It is preferable to make the cover in sections if possible, and secure it in such a manner as will allow it to be easily removed, so that the coils can be taken out and replaced if necessary. Such work, of course, makes it necessary to have sufficient headroom over the brine tank. The same thing applies to the submerged condenser.

1480. Pipe Connections.—After all the principal parts of the plant are in place, the anhydrous-ammonia receiver can be set up. This receiver should always be placed below the level of the condenser, so that the liquid from the latter vessel can drain into it. The oil separator between the compressor and condenser should also be located, and the gauge frame with the pressure gauges should be secured in position. The brine and water circulating pumps, boiler-feed pump, and the rest of the steam-boiler plant should be located and placed according to good engineering practice.

The steam main, from which the pipe to the engine of the compressor branches out, should be provided with a stop-valve to shut off the steam in case of necessity; this in addition to the throttle-valve of the engine. The steam-pipe to the engine should be taken from the top of the steam main, so as to get perfectly dry steam. Bleeders should be provided along the bottom of the steam main to drain any water of condensation that may collect in it, the pipe being pitched in the direction of the bleeder. In large plants, it is advisable to connect with these bleeders a trap which will work automatically and remove all water of condensation from the mains. In small plants, the steam for the pumps is often taken from the under side of the main; this brings all the water of condensation to the pumps, but as these are slow-moving parts with plenty of clearance, there is not the same danger from water as in Corliss engines.

The exhaust steam from the compressor engine can be led into an exhaust main and used for heating, making distilled water, or some other purpose. A valve should be placed in the main exhaust from the compressor engine and one in the exhaust of each pump, to prevent steam backing up into the cylinders when they are not in use or are being packed.

The water should be taken from the water-pump to the ammonia condenser, and there should be a branch pipe to the water-jacket of the compressor. In case it is desired to run the engine as a condensing engine, the overflow water from the ammonia condenser can be utilized as cirlating water for the steam condenser. This is quite often done at a very slight additional first cost and an increased economy of from 15 to 20%.

The ammonia connections consist of a discharge-pipe from the compressor to the oil trap, and one from the oil

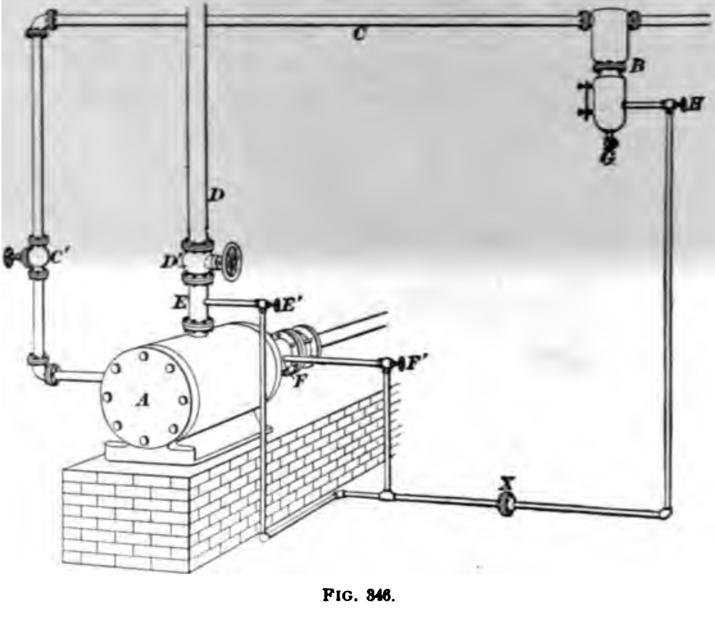


FIG. 346.

trap to the ammonia condenser; a drain-pipe from the ammonia condenser to the ammonia receiver; the expansion-pipe from the receiver to the cooler or expansion tank; a gas suction-pipe from the expansion coils to the compressor; and the gauge pipes commonly known as the "head" pressure and the "back" pressure, these names being given, respectively, to the condensing and evaporating pressure of the ammonia. The head-pressure pipe is usually taken from the discharge-pipe, and the back-pressure from the suction-pipe to the compressor.

Ordinarily the oil from the oil separator is drawn out by means of a cock *G* (Fig. 346) at the bottom of the vessel; though in some systems a connection is made between the oil separator *B* and the gas suction-pipe *D*, and piston-rod stuffing-box *F* of the compressor. By this means a small quantity of oil can be fed into the chamber of the stuffing-box, and if the piston rings begin to groan, some oil can be allowed to enter the cylinder through the suction-pipe. If this arrangement is used, care should be taken to fit the pipe running from the oil separator *B* to the suction-pipe *D* and stuffing-box *F* with a strainer *X* for straining out any impurities, such as scale, that may lodge in the oil separator. The discharge-pipe from the compressor to the condenser should be fitted with an air-valve, or cock, at its highest point, for the purpose of removing air from the condenser. A drain valve should be fitted to the under side of the bottom header of the expansion coils for removing oil from them, and a charging inlet to the suction side of the compressor for taking in anhydrous ammonia for charging the machine. It is also customary to provide an air-valve on the top of the cylinders in single-acting machines and on the discharge-pipe in double-acting machines for the removal of air when the system is being pumped out before charging. In making the suction and discharge connections to and from the compressor, the by-pass connections described in Art. **1404** should not be neglected, as these appliances are indispensable for a successful operation of the plant. Besides the stop-valves on the compressor, a valve should be placed in the discharge main near the condenser, one in the suction-pipe near the expansion coils, and one in the gauge pipes. All pipes and fittings, before being placed in position and after being cut to length, should be well hammered and thoroughly blown out by means of a steam hose for the removal of dirt, scale, etc.

TESTING.

1481. Kinds of Tests.—Refrigerating-machines of the compression type are usually subjected to two classes of tests, viz., the pressure test and the vacuum test. The object of the former is to ascertain the strength of the various parts of the refrigerating-machine and connections, and also the tightness of all joints, etc., and that of the latter is to determine if such joints, together with the packing of the compressor rod, work tight in a vacuum. It often occurs in practice that a piece of mechanism that is perfectly tight under severe pressures leaks in a vacuum.

1482. Testing Water and Steam Pipes.—Before subjecting the machine to the air-pressure test, all the steam, water, waste, and exhaust connections should be tested by turning the live steam in the steam-pipes and subjecting the exhaust-pipes to a slight back pressure, by either partially closing some of the exhaust-valves or by setting the back-pressure valve, if there is one. The water-pipes are usually tested by throttling the inlet water-valves to the condenser and water-jackets of the compressor, whereby a pressure exceeding the ordinary working pressure by about 30% is obtained.

1483. Packing and Inspecting the Compressor.—Before the compressor is started, its cylinder should have a careful inspection, and all grummets which were put on when the plant was being erected should be removed from the various bearings. As a rule, many of the compressors are assembled and run in the shops of the builder. There they are usually subjected to an air-pressure test and all clearances and valves are adjusted; but transportation and subsequent erection often cause derangements, and so it is necessary to examine the different parts of the compressor before the latter is even turned over by hand. The valve covers are first removed from both suction and discharge valves and the cages and valves are taken out. The valve-seats can then be easily seen and examined, and if found perfectly true and clean, are left in place. If, however, they are cut in any way, or if the valves do not lie true in them, they should be removed and ground to a fit, or new seats should be put in. Through the openings of the suction and discharge valves, the inside of the cylinder of the compressor can be seen. It should be carefully examined and cleaned, the cylinder-head being removed if necessary for this purpose. If this is neglected, there is danger of the inner walls of the cylinder being cut before the parts wear down to a bearing. If, however, the cylinder is found to be in comparatively good condition, the compressor should be pinched around by hand until the piston is on the dead-center nearest the removed valve covers. Adjustment for clearance can then be made as described in Art. **1386,** after which the valves and cages can be replaced and the covers can be bolted into place. In case the compressor is double-acting, the clearance at both ends of the cylinder should be examined and the clearance space divided up between them. When the crank is on one of the dead-centers, a mark should be made on the cross-head and also on the guide, for the purpose of identifying the location of the piston in the cylinder in case it is necessary to take up the main crank or wrist-pin bearings.

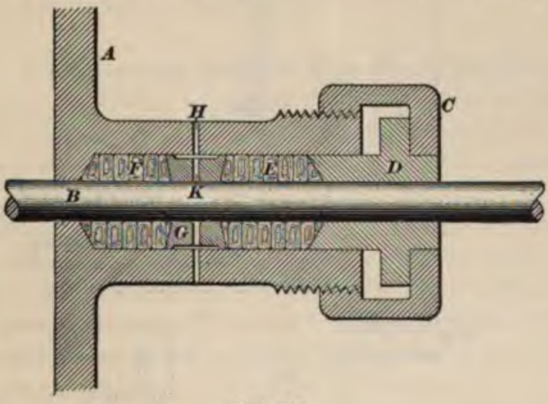


FIG. 347.

The next thing is to pack the piston rod of the engine and compressor. The engine piston rod may be packed

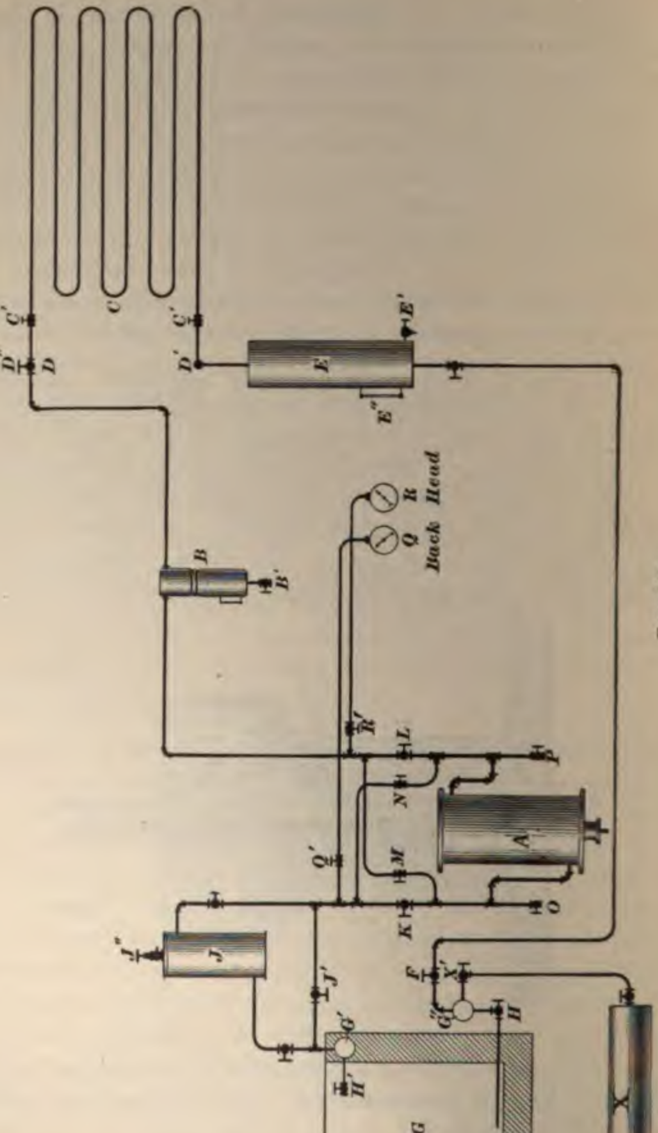


Fig. 348.

with any ordinary good packing. The compressor piston rod, however, should be packed with special ammonia packing. If metallic packing is used, that known as "Common Sense" will be found to give very good results. For a soft packing, Garlock's, Knowlton's, or Crandall's ammonia packing may be used. In packing these compressors, an oil thimble or spool *G*, Fig. 347, is usually provided, which divides the stuffing-box into two parts. This appliance is used more particularly with double-acting compressors. The oil spool *G* is connected with the oil supply of the compressor by the hole *H*, and a certain amount of oil is allowed to enter at this point for the purpose of lubricating the rod. Some of this oil finds its way naturally into the cylinder and serves to lubricate the piston of the compressor.

Seldon's packing has been found very satisfactory on compressors of the wet-compression type. As freezing does not harden or disintegrate it to the same extent that it does packings of a different make, this packing is less likely when frozen to cut the rod.

1484. The Compressor Pressure Test.—After the compressor cylinder has been thoroughly inspected and the piston rods of both engine and compressor packed, the machine is ready for the pressure test of compressed air. This test is made by simply turning the ammonia pump or compressor into an air-compressor; in other words, making the pump suck in air from the engine room and discharge it into the condenser of the ice-machine. Before steam is turned on the engine, however, it is advisable to turn the compressor two or three times by hand, for the purpose of seeing that all parts are in order and do not interfere in any way with one another. The valves of the steam engine should be set according to good engineering practice.

Fig. 348 shows the general arrangement of a compression machine for brine circulation. *A* denotes the compressor, *G* the brine tank, and *C* the condensing coils. The various stop-valves on the pipe, such as *L*, *K*, *C'*, should be inspected and all of them opened, with the exception of the air-valves J', X, E', D'' on the various parts of the apparatus. The main suction-valve K at the compressor should then be closed and the compressor started. It will have no source from which to get air, and after a few revolutions will exhaust the air from the cylinders. The air-cock O on the main suction-pipe is now slightly opened and a little air is admitted to the suction side of the compressor. This will tend to slacken up the speed of the compressor, which is now getting air and doing work in compressing it. After the compressor has been running a few minutes, the head-pressure gauge will begin to indicate an increase of pressure. When the pressure has risen to 75 or 100 pounds, the drain valve E' on the receiver of the condenser should be opened quickly. This will allow the compressed air that has accumulated in the condenser to expand and blow out any dirt, scale, chips, etc., that may have collected in the coils of the condenser. This operation should be repeated several times, and the oil-separator valve B' should be used alternately with E'. When the compressor is first started, the oil feed to the suction and that to the packing should be opened. The water to the jacket should be turned on. It is essential to run the compressor very slowly at the start, and care should be taken to run sufficient water through the jacket to keep the delivery pipe comparatively cool. The object of this is to prevent the oil that enters the cylinder of the compressor from vaporizing. If this vaporization takes place when the machine is worked with ammonia, the oil will condense again in the coils of the condenser, or it may be taken out by means of the oil separator; but when the machine is run as an air-compressor, there is danger of the vapor from the oil combining with the oxygen of the air and forming a very explosive mixture. If, however, the delivery pipe of the compressor is kept cool by running sufficient water through the water-jacket and keeping the speed of the compressor as low as possible, there is little danger of any chemical combination taking place. The West Virginia black oil recommended for lubricating compressor cylinders has a very high fire test, as well as extremely low chill test, and consequently it is not very liable to vaporize in the compressor, even when the latter is used as an air-compressor.

After the coils of the condenser are thoroughly blown out, the expansion-valve *F* is opened wide and the pressure is allowed to rise in the expansion and condenser coils. The back-pressure gauge *Q* should now be shut off, the connection between it and the stop-valve broken, and the stop-valve *Q′* opened quickly several times, so as to blow out the pipe. The same should be done with the high-pressure gauge *R* while the low-pressure gauge is in operation.

After the various parts of the machine, including the condenser coils, expansion coils, oil trap, etc., have been thoroughly blown out, the pressure is gradually increased on the various parts up to 300 pounds, which is the ordinary air-test pressure. When the pressure has reached this limit, one of the assistants should be sent around to inspect all joints, etc.

The air-inlet valve *O* of the compressor is now closed and the machine is shut down; the main suction-valve *K* is opened, thus allowing the full 300-pound pressure to be exerted on all parts of the machine. A careful inspection should be made of the jacket water to see if there are any air-bubbles due to leaks in the flanges of the compressor heads, valve covers, etc. The various joints may now be inspected by painting them with soap-suds. The slightest leak can then be readily seen by means of soap-bubbles. The machine is now allowed to stand for several hours with the test pressure on, the pressure gauge being carefully watched to see if there is any drop. If all the joints are found in good order and the apparatus holds the pressure for a considerable time, the vacuum test may be made next.

1485. Vacuum Test.—The main delivery valve *L*, Fig. 348, is first closed and the air-cock *P* is opened. The various air-cocks in the other parts of the apparatus are opened to relieve the pressure. After the pressure has dropped to that of the atmosphere, all the air-cocks are closed except *P*. The compressor is then started slowly. It will naturally draw the air through the main suction-valve, and as the main delivery valve is closed, will discharge it through the air-cock *P*. Both the back-pressure and high-pressure gauge pipes are now opened, and in a short time the gauges will register a drop of pressure of several inches. The machine is now allowed to run slowly until a vacuum of 30 inches is indicated on both gauges. With a well-constructed machine, it is not difficult to obtain such a vacuum; but with cheap machines having large clearances, the vacuum obtained is seldom more than 26 or 28 inches. As the gauge pressure becomes lower, which indicates a more perfect vacuum, the amount of air discharged from the air-cock on the delivery valve will be less and less; finally, it will reduce to but a slight breadth, and when this point is reached, it will be seen that the compressor ceases to lower the vacuum. The air-cock *P* on the delivery line is now closed, the delivery valve *L* is opened, and the compressor is allowed to stand. If the machine is shut down over night, the gauges should still indicate 30 inches of vacuum on the following morning.

The gauge pipes are then shut off at *Q'* and *R'*, the connections near the gauges are broken, and the pipes to the gauges are filled with oil. The oil protects the gauges and renders their readings more accurate.

CHARGING AND OPERATING.

1486. Charging the Connections.—After the machine has stood the vacuum test for several hours, and all joints, packings, etc., have been found to be perfectly tight, it is ready for charging with anhydrous ammonia and brine. An ammonia drum which has either a $\frac{3}{8}$ or a $\frac{1}{2}$ inch opening is connected with the system at any convenient point of the expansion-pipe or the expansion-coil header *G'*, Fig. 348, by means of the air-valve *X'*. After this connection is made, the compressor is started with the main valves *K* and *L* wide open and the by-pass valves *M* and *N* closed; all other valves are left open except the drier by-pass valve *J′* and the various air-valves, such as *O*, *P*, *D″*, *E′*, etc. The valve on the ammonia drum *X* is then slightly opened, and the pressure in the pipe connecting *X* with the valve *X′* is allowed to rise. If this pipe is found to be tight, the valve *X′* is opened slightly. The ammonia then enters the expansion header *G″*, passes through the expansion coils in the expansion tank *G*, and travels by the various pipes to the compressor *A*. From there it is discharged through the oil separator *B* and stored in the condenser *C*. The gas can go no farther, as the expansion-valve *F* is closed. The feed on the valve *X′* can be so regulated as to keep a pressure of from 10 to 15 pounds on the back-pressure gauge *Q*.

Before the compressor is started, the water should be turned on the condenser *C* and through the jacket of the compressor. Care should be taken that this is always done before starting the machine. With each stroke of the compressor, a certain quantity of ammonia gas is taken from the expansion coil and delivered to the condenser. The pressure in the condenser coils, indicated by the head-pressure gauge *R*, gradually rises until the pressure reached is sufficiently high to condense the ammonia in the coils of the condenser. This pressure depends, of course, on the temperature and quantity of the condensing water. After this pressure is reached, liquid anhydrous ammonia will be seen to accumulate in the gauge-glass *E″* of the receiver *E*. When the ammonia drum, or flask, becomes empty, the pressure on the back-pressure gauge will gradually fall, and the drum will begin to frost on the under side. The valve on the drum is then closed, and after a minute the valve *X′* is also closed; the connection between the drum and *X′* can then be broken, with the loss of but little ammonia. A second drum can then be connected; the valve on it is opened slightly for testing the connecting pipe, then the valve *X′* is opened, and the operations just described are repeated.

1487. Handling Ammonia Drums.—Flasks or drums for anhydrous ammonia are made of steel with heads welded in. They are tested to a heavy pressure, and usually hold one hundred pounds of the liquid. They are made sufficiently large to hold this quantity of ammonia and still leave some room for the expansion of the liquid when subjected to heat. Care should be taken, however, not to place them where they will be exposed to excessive heat, as in the boiler room, or where they can receive the direct rays of the sun. There are several different makes of these drums, each having a different style of connection. The rules for connecting the drums are given by the manufacturer and should be carefully followed by the user.

1488. If at any time it is necessary to remove the ammonia charge from a compression machine, this may be done as follows: An empty ammonia drum is connected with the liquid-anhydrous receiver on the condenser, and the ammonia drum is then packed in ice or snow and set on scales. The cock on the condenser and the one on the ammonia drum are then opened, when the pressure in the condenser will force the liquid into the drum. The latter should be allowed to fill until the scales indicate that about 95 pounds of ammonia have entered the flask. It should then be shut off and another drum connected in its place. As all ammonia drums in common use hold 100 pounds of the anhydrous liquid, there will be room enough left in the drum for the ammonia to expand.

1489. Ammonia Test.—While the pressure is being raised in the condenser, a careful man should be sent to inspect all the connections and note any leaks that the ammonia pressure may show. The water-supply over the condenser should then be diminished so as to raise the head pressure to at least 200 pounds, and all joints should be again carefully examined. If any leaks of a serious nature are discovered, the charging of the machine should be stopped at once by closing the valve X', Fig. 348; the expansion-valve F should then be opened so that the gas may escape into the expansion coil *G*, thus lowering the pressure in the condenser. If the pressure is then sufficiently low to take up the leak, the process of charging can be resumed after the necessary repair has been made. If, however, the leak is so serious as to necessitate the taking out of a fitting or a length of pipe, it is necessary to pump all gas out of the condenser. This is done by closing the two main valves *K* and *L* in the compressor and opening the by-pass valves *M* and *N*. This makes the condenser the suction side of the system and the expansion coils the high-pressure side. The compressor is then kept running until the head-pressure gauge *R* indicates a slight vacuum. The necessary repair can then be made and the expansion-valve *F* opened, by which the pressures in *C* and *G* are again equalized. This pressure will be sufficient in all probability to test the repair.

If, however, no serious leaks have been discovered, the process of charging can be continued until the receiver *E* is about two-thirds full of liquid anhydrous ammonia. The expansion-valve *F* is then gradually opened after the valve *X'* has been closed, and the pressure in the expansion coils is allowed to increase. An experienced man is then sent to examine all the coils and connections on the suction side of the compressor, and a careful test is given these connections, similar to that made for the condenser. A pressure of 150 pounds is usually considered sufficient for the ammonia test on the suction side. After all leaks have been stopped, the machine is ready for service. If the system is of the direct-expansion type, the machine can be started at once on its regular work; but if it is of the brine-circulating type, it is necessary first to charge it with chloride of calcium or with salt.

1490. Brine Charge.—The ammonia test having been successful and the expansion coils having been found tight, the tank is filled about two-thirds full of water and a crib is placed under the return pipe. Into this crib the broken chloride of calcium or the salt is dumped in small quantities. The brine-pump is then started and a stream of water turned on the chloride in the crib. This stream will at the start dissolve the chloride very rapidly until the water in the tank has become well saturated. If the expansion coils remain uncovered after the brine has reached the desired density, sufficient water should be run into the tank to cover them, and enough chloride added to bring the brine to the required degree of saturation. The method of connecting the brine-pump, crib, etc., is shown in Fig. 349.

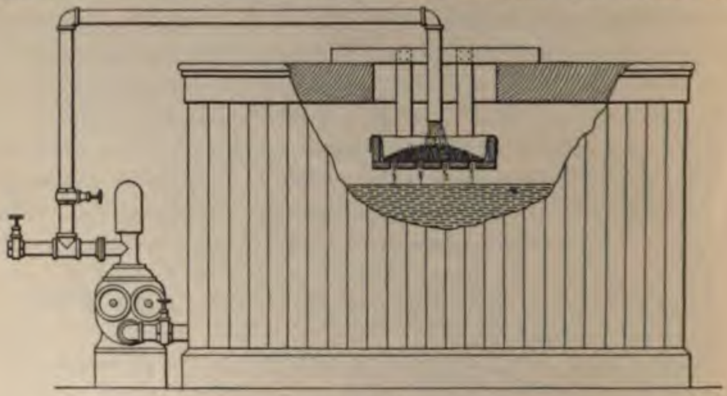


FIG. 349.

1491. Starting the Machine.—Now that the brine charge has been made, the machine charged with ammonia, and all connections tested, the apparatus is ready for service. Before any cooling is possible in the room, it is necessary to cool the brine charge below the freezing point. The brine-pump is stopped, the water-pump is started, and water is run over the condensers and through the water-jacket of the compressor. As the temperature of the brine is probably in the neighborhood of 55°, a comparatively high back pressure can be kept in the expansion coils. As the compressor is started when the pressure in the expansion coils is comparatively high—100 to 150 pounds—it is necessary to reduce this pressure to 30 or 40 pounds before the brine shows any perceptible cooling. The main suction and discharge valves *K* and *L* (Fig. 348) are opened, the by-pass valves *M* and *N* are closed, *F* and *J'* are also closed, and the compressor is started. The back-pressure gauge will begin to indicate less and less pressure. When a pressure of 30 pounds is reached, the expansion-valve *F* is slightly opened, and care is taken to regulate it so as to keep the back pressure between 30 and 35 pounds. After some time the brine temperature will have fallen to about 25°.

The brine-pump can now be started. This will start the circulation in the room and also that in the brine tank, which will help to cool the brine more rapidly. The expansion-valve *F* can be closed a little, so that the back pressure will drop gradually as the temperature of the brine falls. The machine is now in full operation.

If at any time the compressor cylinder begins to groan, some oil should be pumped into the suction-pipe. Care should also be taken that the piston-rod packing is well lubricated; if it begins to heat, the gland on the stuffing-box should be slackened and some oil pumped in. This will cool the rod. It is better to have the piston rod leak a little the first day or two than to have the packing so tight as to cut the rod.

1492. Expansion.—When the brine has cooled to a temperature of 15°, an inspection of the charge should indicate at least 6 inches of liquid anhydrous ammonia in the receiver of the condenser and a back pressure of 20 to 25 pounds. If, however, the back pressure is lower and there is a small quantity of anhydrous ammonia in the receiver, another drum should be connected up and its contents pumped into the machine. As shown in Fig. 348, there is one main expansion-valve *F* and a number of feed-valves *H*, one on each coil, for the purpose of regulating the amount of anhydrous ammonia being fed to the separate coils. These valves *H* should be adjusted once for all, so as to proportion the amount of anhydrous ammonia supplied to each coil; then they should be left alone. The total amount of anhydrous ammonia expanded should be regulated by the main expansion-valve *F*. If the compressor is working properly, the whole action of the machine hinges on the proper manipulation of the expansion-valve. As shown in Art. **1375**, a low back pressure is detrimental to the economical working of a compression machine. Therefore, care should be taken to carry as high a back pressure as possible, and at the same time avoid overfeeding. The best indication of overfeeding is a heavy frost on the suction-pipe of the compressor. Not all of the liquid anhydrous ammonia is evaporated in the expansion coils, and consequently some of it passes over to the compressor. The evaporation in the suction end of the cylinder causes the packing of the piston rod to freeze, and it is liable to leak as soon as it thaws out again. The greatest danger from overfeeding, however, arises from liquid ammonia or oil entering the cylinder of the compressor. The liquid being incompressible, its presence may result in the breakage of a cylinder-head or of a shaft, or the derangement of some part of the compressor. The action in this case is the same as that of a steam engine taking water in with the steam.

1493. Shutting Down the Machine.—To shut down a compression machine, the main throttle-valve of the steam engine is closed, care being taken that the compressor does not stop on a dead-center. The valve on the oil feed is then closed, together with the main suction and delivery valves *K* and *L* and the expansion-valve *F*. The other valves may be left open. The water-pump is then stopped and all the drips are opened; the same is done with the brine-pump. The refrigerating plant is now shut down, and the steam-boiler plant may be shut down as in ordinary practice.

1494. Practical Points Relating to the Starting of the Machine.—After steam is up in the boiler, the first thing to do is to start the water-pump and see that there is a plentiful supply of water going over the condensers. The brine-pump is next started and all its drips are closed. The starting of the brine and water pumps relieves the main steam-pipe of any water that may be in it and prevents the boiler from priming when the compressor is started. The throttle-valve is opened slowly to allow the engine cylinder to heat up, and then the various valves, except the expansion-valve, are opened. After making sure that the water is running over the condenser and through the water-jackets, the compressor is slowly started by opening its throttle. If, however, the engineer is a little awkward in starting the machine and gets it on a dead-center, it is necessary to pry it off center and give it a little lead before the compressor can start off. In case of a big machine, it will be found difficult for one man to pry it over center. The best thing to do is to equalize the pressure on both sides of the compressor piston by closing the main valve *K*, Fig. 348, and opening the by-pass valve *M*. There are notches cast in the rim of the fly-wheel at regular intervals for the purpose of inserting the pinch-bar; they provide an easy means of prying the machine around.

1495. Supplies.—Crude West Virginia black oil, known as Mount Farm oil, is a good lubricant for compressor cylinders, as it has a low cold test, a high fire test, and is inexpensive. Albany grease may be used on the main-shaft bearings and on the other parts of the machine, but it is particularly suited for the low speed and large bearing surfaces of the main shaft. If a composition bearing gives any trouble, the bronze may be cut out and replaced by hard babbit of a good quality. The anhydrous ammonia used should be practically free from foreign substances, among them oil and water. In testing the liquid for impurities, it is preferable to take a sample from a drum that has been connected with the machine and has run nearly empty, as all the impurities, being generally heavier than the liquid ammonia, will then be found in a comparatively small quantity of the liquid.

1496. Operating Details.—A head pressure that is too high is usually due to one of four causes, viz., (1) an insufficient water-supply; (2) too small a condenser; (3) dirty condenser coils; (4) air in the machine. The last-mentioned cause is the most common source of trouble. To remove the air from the condenser, it is first necessary to shut down the compressor. After a few minutes, the air will have risen to the highest point in the system and can be readily drawn off through the valve *D'* (Fig. 348). To do this, run a small piece of pipe from the valve *D'* into a pail of water. As long as large bubbles come to the surface, the gas being drawn out is either air or some impure hydrocarbon, both of which are detrimental to the successful operation of the machine and should be removed. When, however, a sharp, rattling noise is heard, with only a few small bubbles rising to the surface, the valve should be closed, as those are the indications of gaseous ammonia. If the machine is then started, the condensing pressure will be found to have dropped considerably.

The condenser coils should be kept perfectly clean by sweeping them off with a broom. In case it is necessary to use muddy water coming from a stream or brook, these coils should be cleaned at least once every twenty-four hours. Keep the brine charge strong enough to prevent the formation of ice on the coils, since this greatly affects the efficiency of the machine.

ABSORPTION MACHINE.

ERECTION.

1497. Generator.—The generator or retort should be set on a good foundation that will not settle under the weight it has to support. If horizontal, it should be perfectly level. The analyzer should be carefully plumbed after it is put in place. The analyzer trays or plates should then be placed in the analyzer and each tray leveled, so as to insure a good, even surface for the liquor to travel over. The generator should be provided with a pair of gauge-cocks for indicating the quantity of aqua ammonia in it, and all openings, such as gas and liquor connections, should be provided with valves or cocks for closing them. Upon completion, the generator is usually covered with some non-conducting material to prevent radiation.

1498. Condenser and Expansion Tank. — The general rule for the erection of the condenser and expansion tank of the compression machine applies also to those of the absorption type. The only difference consists in the method of connecting the expansion coils, which, as a rule, feed in at the top, the suction being taken from the bottom header. It is the usual practice to put in a gravity connection from the bottom header of the expansion coils to the receiver of the absorber, for the purpose of draining them of any water that may have been entrained with the anhydrous gas and accumulated there.

1499. Exchanger.—The exchanger is usually set on top of the generator, in case this is horizontal, or alongside of it if it is a vertical still. This should be covered, as well as the generator, with some non-conducting material, such as magnesia blocks, asbestos, or hair felt.

1500. Absorber and Aqua-Ammonia Pump.—The absorber should be set as near the expansion tank as practicable, so as to decrease the length of the suction-pipe. It should rest on a good foundation. If the absorber is of the tank type, the foundation should be made similar to that of a submerged condenser; but if it is of the cylindrical type, with either through tubes or helical coils, the foundation should be a good bed of concrete well tamped, so as to prevent the absorber from settling and tipping. It is quite important that the ammonia pump should be placed so that the receiver of the absorber is higher than the suction of the pump; this is to permit the liquid or strong aqua ammonia to run to the pump by gravity instead of being lifted by it. The liquor is charged with ammonia gas, and consequently any lifting action on the part of the pump would create a slight vacuum and liberate the gas; the pump would thus become gas-bound and fail to fill with liquor. The aqua-ammonia pump does not require any elaborate foundation, but merely such a one as an ordinary duplex or single-acting steam-pump needs. A cast-iron bed-plate or pan is a useful addition, as it catches any ammonia or steam that may leak from the pump. Relief-valves should be provided on the caps of the suction and discharge valves, to drain the cylinder.

1501. Rectifier.—The absorption machine of the present day is usually provided with a rectifier, for the purpose of removing any entrained moisture that may pass off from the analyzer on its way to the condenser. The rectifier is usually placed above the analyzer with a gravity drain to the analyzer. Any moisture entrained thus collects in the rectifier and runs through the gravity drain back to the analyzer. To prevent any gas from passing up through the drain-pipe, this pipe is usually provided with a trap or seal. Care should be taken that there is but one trap in this pipe and that the rest of the drip-pipe drains back to the analyzer. The two connecting pipes of the rectifier, viz., the gas line from the analyzer to the rectifier and that from the rectifier to the condenser, should drain, respectively, towards the analyzer and the condenser.

1502. Gauges and Connections.—Pressure gauges should be attached to the generator (or still), to the absorber, and to the expansion coils of the machine. These gauges are steel-spring Bourdon gauges, similar to those used in the compression machine. There should be a valve on the connection between the gauge and the vessel, so that the former may be shut off in case it is necessary to repair it. The generator, condenser, and absorber should be supplied with liquid level gauges, the one on the absorber being usually run open. Air-valves for drawing off air or foreign gases should be located on top of the condenser and absorber and on the absorber receiver. All pipes between the various vessels should have valves, so that any one can be disconnected from the others.

The class of fittings commonly used on the absorption machine are either gland or lock-nut fittings. Soldered-joint fittings are not satisfactory, as the wide ranges of temperature cause the soldered joints to crack. A connection is always run from the bottom of the expansion coils to the receiver of the absorber, so that it is possible to drain the expansion coils into the absorber in case any entrained moisture passes into the condenser and from there to the coils. This connection is usually known as a *blow-out* connection or *purge*.

The generator is connected with the live-steam pipe from the boiler. If there are more coils than one in the generator, they are connected by means of a header, a separate valve being placed on each coil for the purpose of shutting it off separately in case it leaks. The outlets of these coils are also connected to a header with separate valves. This outlet header is usually attached to an ordinary steam-trap, so that the condensed steam passing from the still is discharged by means of the trap. The steam-pipes to the ammonia pump, brine-pump, and water-pump are run as in ordinary practice, and the exhaust is taken to a feed-water heater. Where an absorption machine is used for refrigerating purposes, the steam-trap may be dispensed with, the condensed steam being taken to a boiler-return trap of the Bundy, Pratt & Cady, or Albany type. These traps, being automatic, return the condensed steam to the boiler without requiring any attention on the part of the engineer.

The water connections taken from the water-pump are usually made as follows: In case of excessively warm weather, as in the southern States, a separate supply of water is taken to the condenser, absorber, and rectifier of the machine. If, however, the water is comparatively cool, the condenser and absorber are connected *in series;* that is, the condensing water taken from the condenser is passed through the absorber. This makes a much more economical arrangement of the water consumption of the plant, which in this case is not greater than that of a good compression machine, the amount of water run from the rectifier being about equal to that used over the water-jacket of the compression machine. The waste water from the absorber, condenser, and rectifier is run to the sewer, or, in case of the closed system, it can be discharged into a tank for future distribution.

1503. Coils.—The various coils in the absorber, if they are water coils, should be arranged so that they can be easily blown out with steam or compressed air, to cleanse them of any mud incrustation or scale that may form in them. With comparatively good water, such cleansing is not necessary oftener than once a month, but where river or brook water is used, a frequent blowing out is necessary, especially in rainy weather.

TESTING AND CHARGING.

1504. Cold-Water or Hydraulic-Pressure Test.—As it is impossible to get air-pressure on an absorption machine without the aid of an air-compressor, and as the absorption machine is so constructed and connected that the various parts will drain naturally into the absorber, hydraulic pressure is used instead of air-pressure for testing a machine of this kind. The machine is filled with water by means of the aqua-ammonia pump; then the pump is allowed to run slowly until a pressure of at least 300 pounds is shown by the pressure gauge of the machine. Any leaks are then made tight. The various parts of the machine, however, are usually tested at the shop of the builders to a hydraulic pressure of 500 pounds, the test on the ground being merely for the purpose of determining the tightness of the various connections rather than the actual strength of the machine. Besides, by filling the machine with water, its various parts are thoroughly cleaned.

In order to fill an absorption machine with water, a connection is made such as the one shown in Fig. 350. The main suction-pipe *C* is provided with a tee *G*, from which a temporary suction-pipe *B* is run nearly to the bottom of a barrel *F*. A hose or direct hydrant connection, fitted with a cock *E*, serves to fill the barrel. The valve *C'*

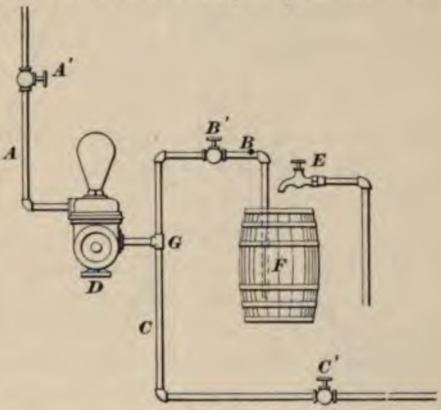
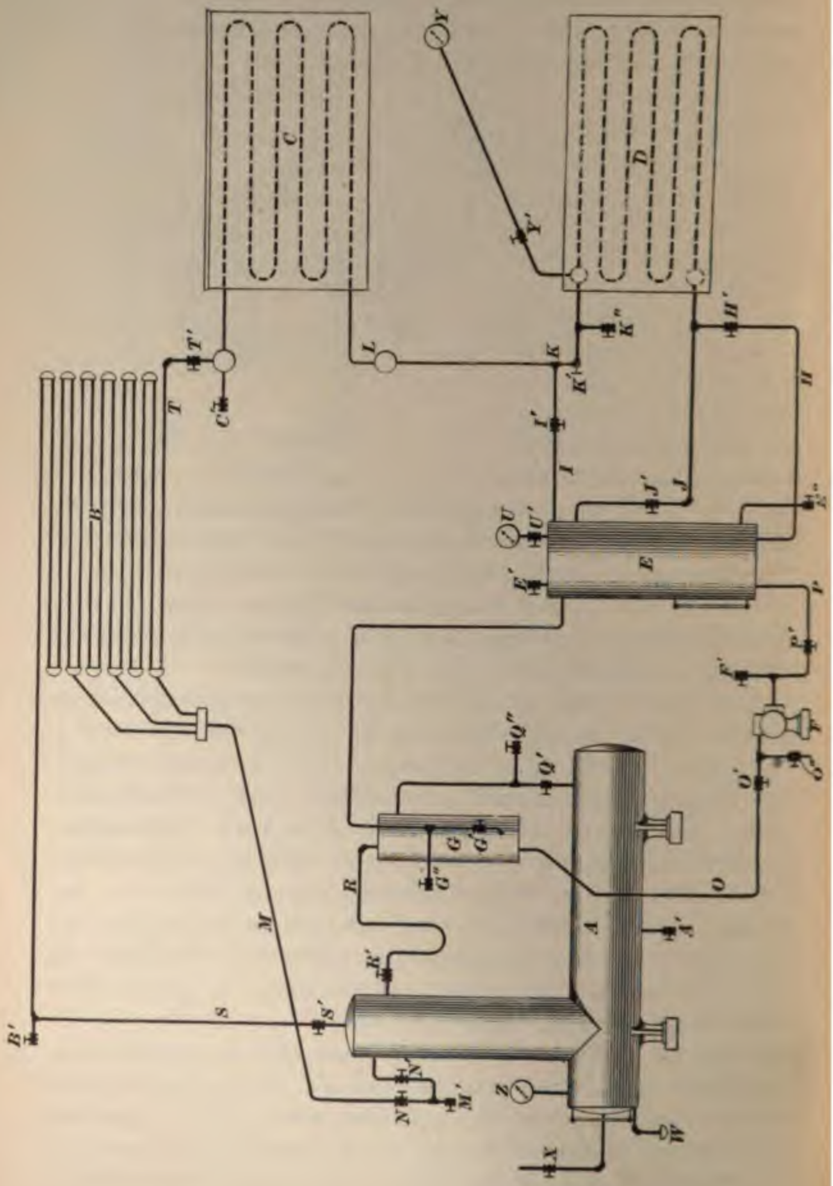


FIG. 350.

is first closed and the valves *A'* and *B'* are opened; then the pump is slowly started and the water is pumped into the machine. The cock *E* is so regulated as to keep the water in the barrel at a constant level. The pump delivering the water into the machine will first fill the coil of the exchanger *G* (Fig. 351), from which the water will flow through the pipe *R* into the analyzer and from there into the generator. Having filled the generator, the water fills the analyzer and rises through the pipe *S* and the drip-pipe *M* into the rectifier *B*. The air-cock *B'* on the rectifier is closed when water is found to run out of it. After filling the rectifier, the water will run to the condenser *C*, from there to the expansion tank *D*, and then to the absorber *E*. When water begins to run out of the upper air purge *E'* on the absorber, the system is completely filled, the exchanger having been filled by opening the valves *Q'* and *G'*. All the air-valves are now closed, and the absorber pressure gauge *U* and the cooler pressure gauge *Y* are shut off by the cocks *U'* and *Y'*. The aqua-ammonia pump is now run very slowly, and with every stroke the pressure will be seen to rise, as shown by the gauge *Z*. When this pressure has reached 150 pounds, a man is sent around to examine all joints, etc., for leaks. If any are found, the pump is shut down, the pressure is relieved by opening one of the air-cocks, and the joint is tightened. After it has been ascertained that all the joints are tight at a pressure of 150 pounds, the pump is again started, and the pressure is gradually raised until the gauge *Z* indicates 300 pounds. As most ammonia gauges are graduated up to only that pressure, it is not advisable to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attempt to raise the pressure any higher. When the pressure has reached this point, the main delivery valve O' is closed, and the pressure is allowed to remain on the machine for at least half an hour. If no leaks are discovered, the machine is ready for the next operation.

1505. Steaming Out.—While the pressure is still on the machine, the gauge connections are broken and the gauges are blown out to clean them. The purpose of steaming the machine is to clean it thoroughly, to heat the joints, so that they can be easily taken up, and to drive out the air. In order to effect this, the following method is usually adopted: The various air-cocks B', C', E', F' (Fig. 351) and the drain-cock A' are opened, and the water is allowed to run out of the machine until the various vessels and the coils in C, D, E, G, and B are empty and the generator A is about half full. The air-cocks are then closed and steam is gradually admitted to the generator coil by means of the valve X; this heats the water in A until it gives off steam. The valve X is gradually opened. The steam generated in A passes up through the analyzer and into the rectifier B, and from there it passes to the condenser C, driving out through the various air-cocks any air there may be in these vessels. The expansion-pipe K, which was broken to drain the condenser of any water that it might contain, is again connected, and the steam is allowed to enter the expansion coils D; from there the steam travels through the gas-pipe J into the absorber E. It will be necessary to carry a steam pressure of about 50 pounds in the generator coil, which will give about 30 pounds pressure in the generator, as indicated by the gauge Z; but by the time the steam gets to the absorber E, there will be but little pressure, owing to condensation and wire drawing. If the valves Q' and G' are opened, the boiling water in A will find its way through the exchanger G into the absorber E, as the difference in pressure between the generator and the absorber will be sufficient to force the water into the latter. The ammonia pump is now started with the main suction-valve P' opened; it will take the condensed steam and hot water from the bottom of the absorber *E* and discharge it into the exchanger *G*, and thence into the analyzer. The pump should be kept running slowly and the water circulating for an hour or more, so as to cleanse the absorber, exchanger, generator, and analyzer thoroughly. The air-valves can be left open on most of the other vessels, and the connection on the expansion-pipe *K* can be partially broken, so that any condensed steam will run out from these various drains. This will insure the expulsion of all air and the washing out of dirt, scale, and other foreign substances.

After the machine has been under steam for several hours and when it does not show any more dirt in the steam, the various air-valves are left open, the absorber is pumped empty, and the water remaining in the generator is blown out from the drain valve *A'*. The flange connection on the line *G'* also is broken, so as to drain the exchanger of any water that it may contain. After all the water is out of the system, steam is still left in the generator coil, which superheats the steam in the generator and prevents any air from entering. When no more steam escapes from the various air-valves, which indicates that the pressure in the machine has dropped to that of the atmosphere, those valves should be closed. The various stop and expansion valves, such as *N'*, *S'*, *B'*, *T'*, *K'*, *H'*, *Q'*, *O'*, and *G'* should be closed, so that after the steam is condensed and a vacuum formed, the air that may leak into any leaky part may not fill the other parts. When the vessels cool down, the various rubber joints should be drawn up, as the heat has softened the rubber and makes it possible to draw such joints up easily. The machine is now ready for charging.

1506. Charging the System.—In order to charge the machine with aqua ammonia, the same arrangement of piping is used as that shown in Fig. 350, but the barrel *F* is replaced by an aqua-ammonia drum. Aqua ammonia comes in wrought-iron glycerine drums holding 750 pounds of ammonia. Each drum has a 2½-inch bung screwed in.

Having removed the bung of one of these drums, a suction-pipe *B* is passed through the bung-hole and is then attached to the elbow of the auxiliary suction-pipe. The pipe *B* reaches to within an inch of the bottom of the drum. As there is a vacuum in the machine, when the valve *B'* in the auxiliary suction-pipe, the delivery valve *A'*, and the valve *C'* in the main suction-pipe are all opened, aqua ammonia will be drawn into the absorber and exchanger of the machine. Care should be taken to sound the drum for the purpose of determining the level of the liquor in it. This may be done by means of a small stick passed down through the bung-hole. By placing one's ear on the suction-pipe near the drum, one can easily tell when the drum is empty by the gurgling sound due to incoming air.

The auxiliary suction-valve *C'*, Fig. 350, is now closed, the pipe entering the drum is detached from the elbow, and another drum is put in place of the first. Then the various stop-valves, such as *S'*, *T'*, etc. (Fig. 351), are opened and as much ammonia is let into the absorber and generator as the vacuum will permit. The main suction-valve to the absorber is then closed and the ammonia pump is started. When the gauge-glass on the generator shows that the latter is full of ammonia liquor, the pump is stopped, the auxiliary suction-valve is closed, and the main suction-valve is opened. The machine is then ready for generating ammonia gas.

1507. The liquid ammonia in the generator is gradually heated by turning steam on through the valve *X*. When the generator is heated to the boiling point of the liquid, the pressure gauge *Z* will indicate from 30 to 60 pounds. As all the main stop-valves are open, this will be the pressure in all the vessels of the machine. Before proceeding with the test, a person is sent around to inspect each joint carefully. If all joints are tight, the steam pressure in the generator coil is allowed to rise, thereby increasing the pressure of the ammonia in the generator. If the temperature of the condensing water, which is now turned on to the rectifier, condenser, and absorber, does not exceed 60°, liquid anhydrous ammonia will be seen to accumulate in the receiver *L* of the condenser when the pressure reaches 120 pounds. After the pressure has reached 60 pounds, as indicated by *Z*, the cocks *U'* and *Y'* are opened and the expansion-valve *K'* is closed. The closing of this valve prevents any of the liquid ammonia from passing into the expansion coils. The valve *Q'* is opened full, and the valve *G'* is opened one or two turns, so as to allow a small quantity of the aqua ammonia, from which the gas has partially evaporated, to pass over into the absorber. When enough liquor has entered the absorber so that it can be seen in the absorber gauge-glass, the ammonia pump is started and kept running at such speed as to keep the level of the liquor in the absorber constant. The valves *J'* and *H'* are now closed. The pressure denoted by *U* will gradually drop to that of the atmosphere, but is not likely to fall below atmospheric pressure, as there will be sufficient air left in *E* to prevent the pressure from dropping below that point. Samples for testing the specific gravity of the ammonia can now be taken at *F'*.

The steam pressure in the generator coil is next gradually increased until it reaches the full boiler pressure. When it has reached that point, the ammonia pump is stopped and the valve *G'* is closed. The machine is now allowed to generate gas from the liquor that is left in the generator. The nearer the steam pressure approaches the condensing pressure, the more complete will be the distillation—that is, the less ammonia gas will be left in the aqua ammonia in the generator. After this process of distilling has gone on for an hour, the specific gravity of the liquor is tested by means of the cooling apparatus shown in Fig. 352, which is attached to the valve *A'*. It consists of a coil *C* of small pipe placed in a bucket *B* of ice-water, and a hydrometer jar *D*. The latter is slowly filled, care being taken that the temperature of the liquor does not exceed 60°. This can be done by running the liquor very slowly out of *A'*. When several samples are taken, the coil *C* should be emptied before taking each sample, so as to obtain the liquor for each test more nearly in the same state as that of the liquor in the generator.

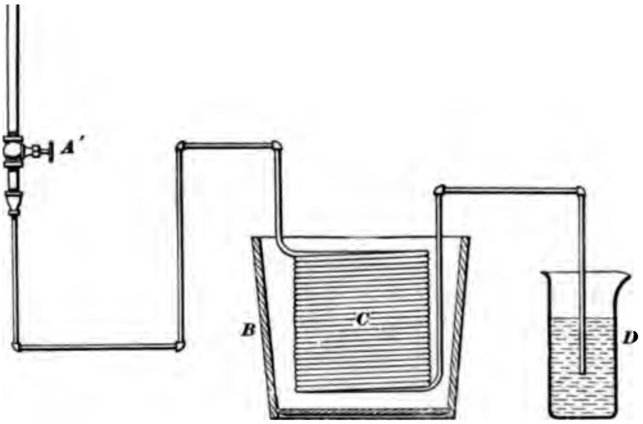

FIG. 352.

Table 30 gives the lowest possible specific gravity to which a solution of ammonia can be reduced under various steam and condensing pressures. Thus, for a maximum steam pressure of 90 pounds and a condensing pressure of 150 pounds, 13½° Beaumé is the lowest possible specific gravity that can be obtained.

TABLE 30.

LOWEST SPECIFIC GRAVITY OF AQUA AMMONIA, IN DEGREES BEAUMÉ.

Steam Pressure in Generator, Pounds.	Ammonia Condensing Pressure, Pounds.					
	100	120	135	150	165	180
60	13.5	15.0	16.0	16.5	17.0	18.0
70	12.5	14.0	15.0	16.0	16.5	17.0
80	12.0	13.0	14.5	15.0	15.5	16.0
90	11.0	12.0	13.0	13.5	14.5	15.0
100		11.5	12.5	13.5	14.0	14.5
120			11.5	12.0	13.0	13.5

[illegible]. After the maximum specific gravity for the given steam pressure has been attained, the weak liquor remaining in the generator is allowed to pass into the absorber. If there is any air remaining in the absorber, its pressure will be shown by the gauge [illegible], which will rise as the weak liquor fills the absorber, thereby compressing the air. When the generator is entirely or nearly empty, as shown by the gauge glass, the valve [illegible] is closed. The liquor that has entered the absorber has now been cooled down so that it can readily be drawn out of the valve [illegible] and allowed to run to waste, or collected in drums, if it can be sold or used for any purpose. In case, however, the air has been all expelled from the absorber *E* (Fig. [illegible]), it will be impossible to run the liquor out of the valve [illegible], as there is a vacuum in the absorber *E*. It is then necessary to start the ammonia pump, closing the valves [illegible] and [illegible] and opening [illegible] and [illegible]. When the pump is started, the liquor will be drawn from the absorber and delivered through the valve [illegible], which can be connected with a sewer or with an ammonia drum.

At this stage an inspection of the different parts of the machine will show the conditions to be as follows: absorber empty, generator with [illegible] inches in the glass, receiver [illegible] anhydrous ammonia. A drum of strong aqua ammonia is now connected with [illegible] and pumped into the generator. This process is continued, fresh drums being connected, until the generator coil is covered with [illegible] inches of strong aqua ammonia. The steam pressure is turned on to the coil, when ammonia gas will be generated from the strong solution in the generator. When the level has dropped [illegible] or [illegible] inches, some weak liquor is allowed to enter the absorber until its level is shown by the gauge glass of the absorber. It will then be found that there is not sufficient aqua ammonia in the generator to cover the generator coil by 1 or 2 inches. It is necessary to keep the generator coil covered to prevent pitting, for the same reason that the tubes of a boiler are kept covered with water.

1509. Charging with Anhydrous Ammonia.—At any time during the summer season, when the machine is found to need more ammonia and can not be shut down long enough to distill off anhydrous from aqua ammonia as described in the previous article, the charge may be strengthened by using anhydrous ammonia. This is done while the machine is in regular operation. The anhydrous drum is connected with the valve K'' (Fig. 351) and the expansion-valve K' is closed. The cock on the drum is then opened wide and K'' is used as a temporary expansion-valve. This allows the ammonia in the condenser to accumulate in the receiver L. After the first drum is empty, a second one is attached, and so on, until the gauge-glass on the receiver L shows that there is a good supply of anhydrous ammonia in the machine. The valve K'' is then closed, the expansion-valve K' opened, and the machine is allowed to feed as before.

1510. Expelling the Air.—Before the machine is ready for work, it is necessary to expel all air and other extraneous gases, whose presence is ascertained by filling the absorber with liquid ammonia, as explained in the preceding article. If, when this is done and the liquor is allowed to cool, the gauge U still indicates a pressure, the air (or other gases) should be allowed to escape through the valve E'. To know when all the air and extraneous gases have been expelled, a small pipe is connected with the purge valve E'', the other end being immersed in a bucket of water. A sharp rattling sound and an absence of bubbles indicate the pressure of ammonia and show that the air and other gases are all out; so long as the latter remain, the water is very much agitated and the bubbles are abundant.

1511. Starting the Machine.—After the air and other gases have been expelled from the absorber, the ammonia pump is started and the steam is gradually turned on to the generator. When a pressure of 120 or 130 pounds has been reached, the valves S' and N' are opened and the water is turned on to the rectifier, condenser, and absorber, These valves should be opened slightly so as not to create too strong a current between the rectifier and the generator, as the difference between the pressures in these two vessels may be considerable. Next, the valve T' is opened slightly. The generator pressure now extends to all the high-pressure parts of the machine. The valves Q' and J' are next opened; the expansion-valve K' is opened a very little until the gauge Y indicates 15 to 20 pounds; then G' is slightly opened, allowing some weak liquor to enter the absorber. As soon as the liquor is seen in the gauge-glass of the absorber, the suction-valve P' and the delivery valve O' are opened and the ammonia pump F is started. The pump is run at such a rate as to keep the liquor in the gauge-glass on the absorber at a constant level. The machine is now doing regular work, cooling the brine in the brine tank D.

1512. Limitations of Capacity.—With an absorption machine well charged, in good condition, and making anhydrous ammonia, there will be found two causes limiting the capacity of the machine: viz., (1) the steam pressure in the generator, and (2) the quantity of liquor circulated between the generator and the absorber. For horizontal generators, the pressure of steam carried on the generator is usually about $\frac{1}{3}$ of the ammonia condensing pressure. For instance, if the condensing pressure is 150 pounds, the steam pressure in the generator will be about 50 pounds. There is no fixed law or rule for machines having vertical stills, and great care must be exercised in not having too strong a charge or too high a steam pressure, or there is danger of "boiling over"—that is, the generation of the gas being very rapid in the still and the liberating area being comparatively small, the whole charge in the still may be blown into the condenser. The quantity of aqua ammonia circulated should not exceed one gallon a minute per ton of ice made per 24 hours, or half a gallon per ton of refrigerating effect.

Having started the ammonia pump at the proper speed, the weak-liquor valve G' is so regulated that the quantity of ammonia, as shown by the absorber gauge-glass, will remain constant. The amount of feed depends largely upon the frost on the suction-pipe J. When the brine is cooled down to 15° or 20°, this frost should come close to the absorber, without, however, entering the absorber itself.

EXAMPLE.—An absorption machine has a double-acting ammonia pump, the cylinder of which is 2½ inches in diameter and the stroke is 7 inches. The capacity of the machine is 10 tons. How fast should the pump run?

SOLUTION.—The number of gallons of ammonia circulated per minute is $\frac{1}{2} \times 10 = 5$, which is equal to $5 \times 231 = 1,155$ cubic inches. The volume of the cylinder $= \frac{\pi}{4} \times (2\frac{1}{2})^2 \times 7 = 34.36$ cu. in. As this is the quantity of ammonia pumped per stroke, the number of strokes necessary to pump 1,155 cubic inches, that is, the number of strokes per minute, is $\frac{1,155}{34.36} = 33.6$. The number of revolutions per minute is half of this, or 16.8. Ans.

1513. Defective Working.—If, after the machine has been running some time, the pressure on the gauge Y (Fig. 351) gradually falls, while the temperature of the brine remains constant, this is an indication that the machine is not making anhydrous ammonia. The usual remedy for this is to run more water over the rectifier coil B, so that more ammonia will condense in the coil and return to the analyzer through the line M.

Another indication of poor working is the fall of the level of the liquid in the generator while remaining constant in the absorber. This means that some of the aqua ammonia in the generator is working over into the condenser, and from there into the cooler expansion coils, where it remains. In order to drain out these coils, the valve H' is opened slightly until the pipe H is frosted all over. This valve should not be opened too much, or the ammonia pump F will begin "kicking" on account of the liquor getting too strong in the absorber; but if opened slightly, the liquor will gradually drain out of the expansion coils and be carried over into the generator by means of the aqua-ammonia pump.

QUANTITY OF WATER USED.

1514. *Rectifier.*—The amount of water delivered to the rectifier should be governed by the temperature of the pipe *T*, which is kept from 15° to 20° higher than that of the condensing water.

Condenser and Absorber.—Sufficient water should be run over the condenser to keep the head pressure as low as possible, and enough over the absorber to keep the outlet water below 110°. If, however, it is not possible to get enough water to produce a sufficiently low temperature, a water-cooling tower or gradier may be installed with economy, as the fuel consumption will increase as the quantity of water is decreased. In a well-designed absorption machine, two gallons of water at 60° per ton of refrigerating effect, or four gallons per ton of ice made, is usually sufficient. This quantity must be increased as the temperature of the water increases.

Weak-Liquor Cooler.—If either the absorber or the exchanger of a machine is too small, and on this account the weak liquor entering through the exchanger *G* is very hot, a cooling coil may be inserted between the exchanger and the absorber. The weak liquor from the exchanger enters the bottom of this coil, the outlet of which at the top is connected to the absorber. Water is allowed to trickle over this coil, the arrangement being similar to that of an atmospheric condenser. Such a coil will increase the capacity of the machine from 10 to 15%.

POINTS ON OPERATING.

1515. Specific Gravity of Liquor.—In the majority of absorption machines, the weak liquor leaves the generator at from 18 to 20 degrees Beaumé, and the strong liquor leaves the absorber at from 25 to 30 degrees Beaumé. The specific gravity of the liquor in the generator depends upon the steam pressure, the condensing pressure of the ammonia, and the amount of steam-coil surface; that of the liquid in the absorber upon the temperature and quantity of the condensing water, the surface in the absorber, and the back pressure. Oftentimes the specific gravity of the strong liquor is taken in order to ascertain whether the machine is sufficiently charged. But such specific gravity does not indicate the strength of the charge. For example, if with a heavy charge in the machine and a light feeding at the expansion-valve, the back pressure is made to run low, the specific gravity of the liquor in the absorber becomes correspondingly low. If there is plenty of water passing through the absorber, a good charge of aqua ammonia in the machine, and the strong liquor is not so strong as it should be, the cause is probably either air or foul gas in the machine or dirt in the coils of the absorber. These coils should be kept clean by blowing them out occasionally with steam or compressed air, the latter being preferable if available. If very hard water is used, such coils should be blown out at least once a week. After giving the absorber a thorough cleaning and purging it free of air, the specific gravity of the strong liquor will be found to have increased by several degrees.

1516. Leaky Coils.—A slight leak in the generator, condenser, absorber, or exchanger coils, although it may not be discovered on inspection, causes a falling off in the capacity of the machine and a loss of ammonia. A leak in the coils of the generator is easily detected by opening the vent valve on the distilled water leaving the generator coil, when the odor of gaseous ammonia will be noticed, as all the ammonia will vaporize in coming in contact with the air, owing to the high temperature of the latter. Another way is to cool some of the condensed steam from the generator coils and apply the Nessler reagent test.

To determine a leak in the condenser or absorber coils, draw some of the overflow water from either one of these vessels and then apply the Nessler test. In case the absorber has water coils, a sample of water should be taken from each coil independently and tested. Then the leaky coil should be capped or plugged and the machine allowed to run with the other coils until the water cools off sufficiently to permit the machine being shut down for replacing the defective coil.

A leak in the exchanger coil is not so easily detected, particularly as there is no loss of ammonia noticed in the machine, the only loss being in the capacity. To test this coil, a tee with a valve Q'' (Fig. 351) is placed in the weak-liquor pipe between the generator and exchanger, and another one at G'' in the weak-liquor pipe between the exchanger and absorber. A cooling coil (see Fig. 352) is then connected with both of these valves. The specific gravity of samples of ammonia taken from one of the valves should be the same as that of samples taken from the other. If the sample from G'' shows a higher specific gravity than that from Q'', there must be a leak in one of the exchanger coils.

HOW TO REDUCE THE PRESSURE IN ANY VESSEL.

1517. Absorber.—The absorber is to the absorption machine what the suction of the compressor is to the compression machine. It is that part of the absorption machine by which a pressure may be reduced almost instantaneously, if the machine is in good running order and the absorber free from air or foul gas. By shutting the valve J' (Fig. 351), the absorber pressure should drop to 10 or 15 inches of vacuum inside of 5 minutes. It will therefore be seen that if it is desired to reduce the pressure in the absorber, all that is necessary is to close the valve J' and pump out the absorber the same as in ordinary running.

1518. Expansion Coils or Cooler.—If it is desired to reduce the pressure in the expansion coils or cooler, the expansion-valve K' is closed, and after the machine has been running for half an hour the drain-pipe valve H' is opened so as to permit any liquor that is left in the expansion coils to pass into the absorber, the valve J' being closed. While this has been going on, the machine will have been generating anhydrous ammonia, which is stored in the condenser.

1519. Condenser.—It is very convenient to make a by-pass connection *I* between the absorber and the expansion-pipe *K*. If it is desired to make a vacuum in the condenser coils, all the anhydrous ammonia is fed into the expansion coils by opening *K'* wide and closing *T'* until the pressure in the condenser coils has been reduced to that of the expansion coils, viz., 15 to 20 pounds. The expansion-valve *K'* is then closed, the by-pass valve *I'* opened, and *J'* and *H'* are closed. The gas that remains in the condenser will then be drawn into the absorber, and the pressure in those parts will be reduced.

If it is desired to reduce the pressure in the rectifier, the same thing is done, except that the valve *T'* is left open and the valves *S'* and *N'* are closed.

In case there is no by-pass connection on the machine, the steam is shut off the generator, and when the condensing pressure drops slightly, the valves *N'* and *S'* are closed, the other valves being left open as in regular running. The pressure in the rectifier, condenser, cooler, and absorber is then gradually worked down, the anhydrous ammonia being gradually stored in the generator. This, however, is a very tedious operation, it being much quicker to run all the anhydrous ammonia from the condenser into the cooler, and then, when the condenser is under cooler pressure, open up the cocks *C'* and purge the condenser of the remaining pressure. It will be found, as a rule, that only about one-half of the pressure left in the condenser is due to ammonia, the other half being due to air. Consequently, the loss of ammonia is not very great.

1520. Generator.—To reduce the pressure in the generator, turn on a high steam pressure so as to distill off as much ammonia gas as possible, then close the valve *T'* and shut down the machine, leaving the valves *S'* and *N'* open and water running over the rectifier coil; break the steam connection to the generator coil and run a water hose inside of this coil, turning on a very small stream of water. Care should be taken not to chill the steam coil too rapidly, as it is then liable to crack or be strained. The pressure in the generator will be found to drop very rapidly, and in about an hour the generator will be accessible for repairs.

1521. Ammonia Pump.—The suction and delivery valve chambers of the ammonia pump should be fitted with vent-cocks or valves, so that they may be emptied whenever it is desirable to examine the condition of the pump-valves. When this is being done, the main suction and delivery valves *P'* and *O'* should be closed, and the liquor from the suction and discharge chambers of the pump drawn out through the vent-cocks into a pail of water. The caps covering the valves can then be taken off and the valves examined. If trouble is found in keeping the piston rod of the ammonia pump packed tight, a stuffing-box arranged like that shown in Fig. 347 can be put on the pump. The spool should have a bore equal to the diameter of the rod, or a trifle larger, and the outside diameter of the spool should be $\frac{1}{64}$ inch less than the inside diameter of the stuffing-box. The length of this spool should be from 1 to $2\frac{1}{2}$ inches, according to the diameter of the rod. Care should be taken that the spool is drilled with several holes (marked *K*), so that any leak along the piston rod will easily pass up through these holes to the outer circumference of the spool. The stuffing-box is drilled and tapped at about one-half its length for $\frac{1}{4}$ or $\frac{1}{2}$ inch pipe at *H*. This pipe connects with the suction side of the pump. All that is necessary in this case is to pack the portion *E* between the spool and the gland, so that it will hold against the suction pressure. The portion *F* that has to hold against the discharge pressure is better packed with a packing having very little friction. Though this packing may not be perfectly tight, any leakage will pass along the rod and through the holes *K* to the suction side of the pump, through the pipe connection at *H*.

APPLICATIONS OF REFRIGERATION.

1522. The principles and methods of refrigeration explained in this Course find their most important application in the manufacture of beer, the preservation of perishable articles (especially victuals) in cold-storage rooms or boxes, and the production of artificial ice.

BREWERY REFRIGERATION.

GENERAL DESCRIPTION OF THE PROCESS.—PROPERTIES OF WORT.

1523. How Beer Is Made.—The making of beer consists of three different operations, viz.: (*a*) the preparation of malt from barley; (*b*) the preparation of wort from malt; (*c*) the fermentation of wort to convert it into beer.

By washing specially prepared malt with hot water, a dilution is obtained known as *clear wort*. This is boiled with hops in a copper vessel, and the resulting product, which is the *beer wort*, or simply *wort*, is first cooled and then converted into beer by adding yeast, which causes the chemical decomposition of the wort. This decomposition is called *fermentation*, and is effected in large vessels placed in rooms cooled by refrigerating machinery.

1524. Properties of Wort.—Wort consists mainly of saccharine and dextrine dissolved in water. The **strength** of the solution is measured by the amount of solid matter (that is, dextrine and saccharine) it contains. It is determined by a kind of hydrometer known as a **Balling saccharometer,** so graduated that, when immersed in the liquid, it registers the percentage of solid matter the liquid contains.

The following table gives the specific gravity and the specific heat of wort of different strengths:

TABLE 31.

SPECIFIC GRAVITY AND SPECIFIC HEAT OF WORT AT 60° F.

Strength, by Balling Saccharometer.	Specific Gravity.	Specific Heat.
8	1.0320	.944
9	1.0363	.937
10	1.0404	.930
11	1.0446	.923
12	1.0488	.916
13	1.0530	.909
14	1.0572	.902
15	1.0614	.895
16	1.0657	.888
17	1.0700	.881
18	1.0744	.874
19	1.0788	.867
20	1.0832	.861

For any other temperature t, the specific gravity s is given by the formula

$$s = s_{60} + .00015\,(60 - t), \qquad \textbf{(129.)}$$

in which s_{60} is the specific gravity at 60°, as given in the table. If t is greater than 60, the factor $60 - t$ is negative, which means that $t - 60$ should be multiplied by .00015, and the result subtracted from s_{60}.

EXAMPLE.—What is the specific gravity of a wort of 18% strength. (*a*) at a temperature of 45°? (*b*) at a temperature of 70°?

SOLUTION.—(*a*) From the second column in the table we find $s_{60} =$ 1.0744. Here $t = 45$, and formula **129** gives

$$s = 1.0744 + .00015\,(60 - 45) = 1.0744 + .00015 \times 15 = 1.0767. \quad \text{Ans.}$$

(*b*) Here $t = 70°$, and $60 - t = -10$. The same formula gives $s = 1.0744 - .00015 \times 10 = 1.0729$. Ans.

THE COOLING OPERATIONS.

1525. Ends to be Attained.—In the refrigeration of breweries, there are three things to be accomplished, viz.: (1) cooling the wort, (2) removing the heat of fermentation, and (3) cooling off the cellars, fermenting rooms, etc., to a temperature of between 34° and 38° F.

1526. Cooling the Wort.—The wort is taken from the brew kettle at a temperature of about 200° and drawn into large shallow iron tanks *C* (Fig. 353) open to the air, where it cools to a temperature of about 110° in from 2 to 3 hours. From there it is allowed to trickle over a **Baudelot cooler,** which consists of two coils *A* and *B*, one under the other, kept at a low temperature in order to cool the wort. Water at a temperature of about 60° is kept circulating in the upper coil *A*. The lower coil is kept at a still lower temperature by the circulation of either ice-water or brine,

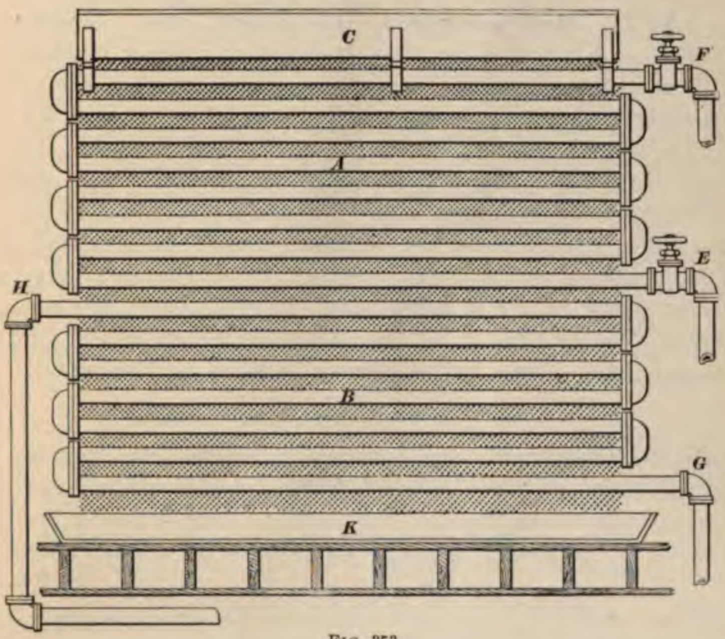

Fig. 353.

taken from the brine tank through the pipe G and returned to the same tank through a pipe H. Another way of cooling the coil B is to connect it with the refrigerating-machine through the pipe G and use it as an expansion coil. When, after running over the two coils of the cooler, the wort reaches the pan K, where it is collected, its temperature is reduced to about 40°.

When brine is used in the Baudelot cooler, the coils are usually made of copper tinned over. If the lower coil is used as an expansion coil, it is made of polished steel pipe.

1527. Refrigeration Required for Cooling Wort.—The capacity of a brewery is usually expressed by the number of 31.5-gallon barrels brewed per day of 24 hours. Let this number be denoted by b. Also, let g be the specific gravity of the wort, s its specific heat, T its temperature after leaving the upper coil of the cooler, t the temperature to which it must be reduced in trickling over the lower coil, and H the number of B. T. U. required to effect this reduction of temperature. The value of T may be taken equal to the temperature of the water in the upper coil. The weight of 31.5 gallons of water being 262.4 pounds, that of one barrel of wort is 262.4 g. Therefore,

$$H = 262.4\ g\,b\,s\,(T - t). \qquad \textbf{(130.)}$$

If F is the number of tons of refrigeration corresponding to this heat, we have (formula **109**)

$$F = \frac{H}{285{,}300} = \frac{262.4\,g\,b\,s\,(T - t)}{285{,}300} = .0009197\,g\,b\,s\,(T - t). \qquad \textbf{(131.)}$$

The values of g and s are taken from tables, assuming a mean temperature $= \frac{1}{2}\ (T + t)$. Taking $T = 70°$, $t = 40°$, $g = 1.05$, and $s = .91$, the preceding formula becomes

$$F = .0009197 \times 1.05 \times .91 \times 30\ b = .02636\ b = \tfrac{1}{38}\ b, \text{ nearly.} \qquad \textbf{(132.)}$$

If in this formula we make $F = 1$, we get $b = 38$; that is, 1 ton of refrigerating effect will cool 38 barrels of wort from 70° to 40°.

EXAMPLE.—(*a*) How many barrels of wort can be cooled in 24 hours from 70° to 40°, the refrigerating effect being 6.5 tons? (*b*) What must be the refrigerating effect of a cooler that it may cool 275 barrels of wort a day from 75° to 40°?

SOLUTION.—(*a*) From formula **132** we get

$$b = 38\,F = 38 \times 6.5 = 247 \text{ barrels. Ans.}$$

(*b*) Taking $g = 1.05$ and $s = .9$, formula **131** gives

$$F = .0009197 \times 1.05 \times 275 \times .9\,(75 - 40) = 8.4 \text{ tons. Ans.}$$

1528. Heat of Fermentation.—After leaving the Baudelot cooler, the wort is pumped into large vats or tanks placed in the fermenting room, where yeast is added to it in order to start and keep the process of fermentation. This room is kept at a temperature of about 40°. As the wort in the tubs ferments, it gives off a certain amount of heat. This heat is removed by means of **attemperators,** which are coils of iron pipe suspended in the fermenting tubs and kept at a low temperature by cooled water or brine made to circulate in them. The diameter of each coil is about two-thirds the diameter of the fermenting tub. The coils should have enough turns to give about 12 square feet of cooling surface for every 100 barrels of wort.

1529. Refrigeration Required to Remove the Heat of Fermentation.—The number H of B. T. U. necessary to dispose of the heat of fermentation of n barrels of wort is determined by the following formula:

$$H = 3\,n\,(s - s')\,(259 + s). \qquad \textbf{(133.)}$$

In this formula, s is the strength of the unfermented wort and s' the strength of the fermented wort (the beer), both as given by a Balling saccharometer. The corresponding number F of tons of refrigeration is

$$F = \frac{3\,n\,(s - s')\,(259 + s)}{285{,}300} = .00001052\,n\,(s - s')\,(259 + s). \qquad \textbf{(134.)}$$

A approximation, which is sufficiently close for many oses, as for general estimates, is obtained by taking and $s' = 4$, in which case the value of F becomes

$$F = .028\,n = \frac{n}{36}, \text{ nearly.} \qquad \textbf{(135.)}$$

If in formula we make $F = 1$, we get $n = 36$, which hows to remove the heat of fermentation 1 ton of refrig d for every 36 barrels of beer. Th str eers. For weak beers, 1 ton of refrigera ity s ffice for 50 or 60 barrels. For rough and p nary estimates, 25 barrels are usually allowed per ton of refr erating capacity.

1530. Cooling the Cellars.—After the process of fermentation is completed, the beer is drawn off from the various fermenting tubs into the storage cellars, where it is ready to settle and age. These cellars are cooled by means of coils of pipe placed on either the ceiling or the side walls, and whose temperature is kept down either by using them as expansion coils or by allowing a current of brine to circulate through them. The amount of refrigeration required for this purpose depends greatly on the manner in which the rooms are insulated, as will be explained presently. With the ordinary insulation generally used, 1 ton of refrigeration may be allowed for every 10,000 to 14,000 cubic feet of storage room, where the temperature is kept at between 32° and 38°. This applies to large breweries. In breweries of 50,000 cubic feet or under, about 5,000 to 7,000 cubic feet per ton of refrigeration should be allowed.

INSULATION OF ROOMS.

1531. Object of Insulation.—The object of insulating a cold-storage or freezing room is to prevent it from taking in heat, either by radiation, by conduction, or by convection. The less the heat is that enters a room from the outside, the less will be the work required to cool it.

Consequently, if a room is well insulated, a smaller refrigerating capacity is required to keep it at the necessary temperature than when the room is poorly insulated. This means that less fuel will have to be expended to cool the space. The economic importance of insulation is therefore manifest.

1532. Insulators Used.—For the purpose of insulation, rooms are divided into damp rooms and dry rooms. Brewery refrigerating rooms and ice-storage houses are examples of the former class of rooms; while cold-storage and freezing warehouses would come under the latter class.

The ordinary insulating materials are wood and paper. The latter, however, is not well adapted to the insulation of damp rooms, where it is not durable. In late years this class of rooms has been insulated largely by means of hollow brick walls containing from one to three air-spaces. To make the air-spaces more perfect, the walls are covered with a coat of either pitch or paraffin wax. This method of insulating is found to be very efficient for comparatively high temperatures, such as are required in breweries and packing houses, and is not nearly so liable to contract mold or mildew as wood is.

Dry-storage rooms have always been insulated with wood and paper, with air-spaces, or air-spaces filled with some filling material, such as sawdust, planing-mill shavings, mineral wool, cork, wood-ashes, cinders, etc.

1533. Air-Spaces.—The best and cheapest non-conductor is air, but in order to make it efficient, it is necessary to make a "dead air-space," that is, so to inclose the air on all sides as to prevent its motion; otherwise there would be air-currents, by which a great deal of heat would be conveyed from the outside to the rooms. For this reason, brick walls are pitched or covered with paraffin, and when wood is used instead of brick, paper is laid between boards so as to prevent the escape of the air confined in the air-spaces.

In making an air-space with boards, it is always best to use double boards with paper in between, the double boarding always be easily applied, thereby thoroughly cleansing the walls and sweetening the atmosphere of the room.

In buildings where iron construction is used, the wrought-iron or hollow cast-iron columns which pass through the rooms should be insulated very carefully to prevent any conduction.

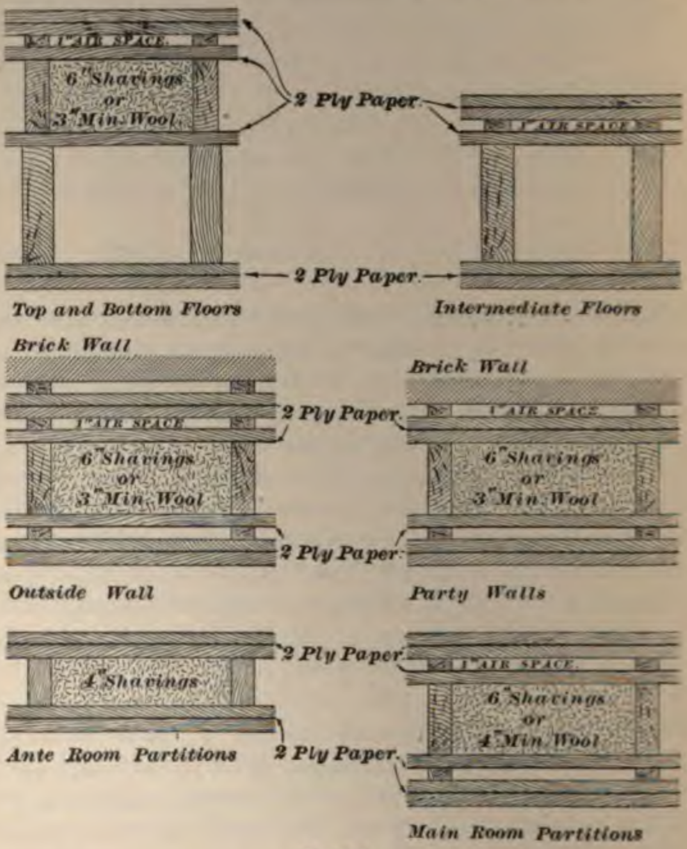

FIG. 354.

In all the foregoing illustrations it will be noticed that in case of brick walls an air-space is generally made by tacking strips to the wall. The object is to allow any moisture that may accumulate on the wall to run down the face of the wall all the way through to the cellar, without injuring the insulation of the building. It is advisable to have a sill or plate at the foot of the insulator, around the walls of the room, to keep the insulation from getting wet when the floor is washed.

1538. Insulation of Exposed Surfaces.—Exposed pipes or other surfaces through which a refrigerating medium is passing will soon be covered with frost. This is due to the moisture in the atmosphere being deposited on the pipe and then frozen. To thoroughly insulate such surfaces, it is not only necessary to keep the heat from passing through them, but to keep the air out; in other words, to make the insulation perfectly air-tight. It is also necessary to have the insulation consist of such substance as will not readily be injured by moisture. Hair felt was largely used in former years; this was wrapped around the pipe and then covered with paper; then more felt and more paper, and so on for several layers; then the whole was sewed in a canvas envelope and covered with several coats of good water-proof paint. Such insulation is found to be comparatively efficient and will last several years, but eventually the air will work through the hair felt, and then the pipes will begin to sweat, and if left long enough, freeze.

The "Voorhees" covering is a much better insulation. It is made by soaking the hair felt in boiling resin and is applied to the pipe while the resin is still hot enough, so that the sheets of felt can be easily molded to the shape of the pipe. These sheets are tied fast with heavy twine; two layers of the covering are applied, and after the resin has cooled and set the string is cut off. This insulation is probably the best on the market, but is very expensive, except in large quantities, as it is necessary to prepare it on the spot.

The best sectional insulation is the Nonpareil Cork covering, which consists of granulated cork soaked in a composition and molded up in the form of sectional insulation similar to magnesia sectional steam-pipe covering. Two layers of this are applied so as to break joints, and the whole is given a coat of rubber cement, which makes it practically air-tight.

In case of large brine mains that are boxed or laid under ground, a mixture of ground cork and pitch is found to give very satisfactory results. The pitch is first melted, then the cork is mixed with it, and the mixture is poured over the pipes and well tamped around them.

COLD STORAGE.

COLD-STORAGE WAREHOUSES.

1539. Object of Cold-Storage Warehouses.—Cold-storage warehouses are used for the preservation of perishable goods by means of a low temperature, which prevents decay. The goods are placed in rooms, and there kept sometimes for months, separate rooms being usually necessary for different kinds of goods, not only because different substances often act on and affect one another, but also because, as experience shows, they require different temperatures for their preservation.

1540. Conditions to be Maintained. — For the proper preservation of goods, it is necessary (1) that the air should be often renewed; (2) that the air should have the proper amount of moisture; (3) that the temperature should remain within certain limits.

The first requisite is obtained by a proper system of ventilation, to be described later; the second by a careful use of the hygrometer and psychrometer for ascertaining the relative humidity of the air, care being taken not to have the air too dry, as this may result in the evaporation of the goods, nor too damp, as this will cause mold or must. The amount of moisture that air can contain increases with the temperature. If when the air enters the room it is very damp, some of the moisture will be precipitated as the temperature falls, and adhere to the pipes, where it will freeze. Care should be taken to keep the pipes as free from this frost as possible.

The temperature is controlled by the refrigerating-machine. For regular public cold storage, the brine-circulation system has in general been found somewhat superior to the direct-expansion system. The latter has often been tried, but has been abandoned in some cases, owing to the risk of the ammonia leaking and to the difficulty in maintaining a steady temperature.

1541. General Design.—The majority of public cold-storage warehouses are built of brick or stone and preferably of the slow-burning mill construction, the best design being that of the perfect cube, as this gives the least exposed wall and roof area for the greatest cubical contents. In order to avoid the heat from the boiler and engine, they are placed in a separate building adjoining the main storage building. As a rule, the freezing rooms are placed together on the lower floors of the building, the cellar being often used for keeping frozen fish, etc. Such rooms have concrete floors, and the ground under these floors is frozen through for several feet, owing to the constant low temperature maintained in the rooms.

It is bad practice to place the freezing rooms above cold-storage rooms, as, in spite of the best insulation, the ceiling of the cold-storage rooms will sweat, owing to the cold striking through the floor of the freezing room and the moisture in the air of the cold-storage room naturally condensing on these cold surfaces.

REFRIGERATION REQUIRED.

1542. The refrigeration required for cold-storage rooms may be divided into two parts, viz.: (1) that required to keep the room at the required temperature, by preventing radiation through the walls; (2) that required to cool the articles brought in the room from their temperature to the temperature of the room.

1543. Refrigeration Required to Keep the Temperature of the Room.—The refrigeration required to keep the room at a constant temperature is computed by means of the following general formula:

$$H = c\,A\,(t - t_1), \qquad \textbf{(136.)}$$

in which H = B. T. U. of refrigeration required to maintain a given space at a certain temperature t_1, when this space is separated from another, in which the temperature is t, by a surface whose area in square feet is A; and c = constant depending upon the material and thickness of the substance separating the two spaces. The value of the constant generally varies between 2 and 5. For rough estimates, it may be taken as equal to 3.

The preceding refrigeration may be reduced to tons by dividing by 285,300; that is, the amount F of refrigeration, expressed in tons, is

$$F = \frac{c\,A\,(t - t_1)}{285{,}300} = .000003505\,c\,A\,(t - t_1). \qquad \textbf{(137.)}$$

1544. Table 33 contains values of c as given in Siebel's *Mechanical Refrigeration*.

For double floors and ceilings, air-tight and well filled, so as to prevent the ingress of air, c may be taken as 2. When a room is separated from the outside by a hermetically closed air-space between two walls, the value of c for the outside wall may be used in the formula, but instead of the temperature t_1, a mean should be taken between t_1 and t, which is equivalent to using $\frac{1}{2}(t - t_1)$ instead of $t - t_1$.

For a wall consisting of several materials, the coefficient c may be found from the formula

$$c = \frac{1}{\dfrac{b_1}{c_1} + \dfrac{b_2}{c_2} + \dfrac{b_3}{c_3} + \ldots}, \qquad \textbf{(138.)}$$

in which b_1, b_2, b_3, etc., are the thicknesses, and c_1, c_2, c_3, etc., the corresponding values of c for the several materials composing the wall.

In large cold-storage warehouses of 250,000 cubic feet or over, 1 ton of refrigeration will maintain 10,000 cubic feet of well-insulated space at a temperature of 30°, and 5,000 cubic

TABLE 33.

VALUES OF THE COEFFICIENT c IN FORMULA 137.

Partition.	Thickness, Inches.	c.
Single Windows		12.0
Double Windows		7.0
Pine Wood	12	2.0
Mineral Wool	12	1.6
Granulated Cork	12	1.3
Wood-Ashes	12	1.0
Sawdust	12	1.1
Charcoal (powdered)	12	1.3
Cotton	12	0.7
Soft Paper Felt	12	0.5
Brick	4½	5.5
"	9	4.5
"	14	3.6
"	18	3.0
"	27	2.6
"	36	2.2
Stone (masonry)	6	6.2
"	12	5.5
"	18	5.0
"	24	4.5
"	30	4.3
"	36	4.1

feet at 15°. In small warehouses of 50,000 to 100,000 cubic feet capacity, 1 ton of refrigerating effect will maintain 6,000 cubic feet at cold-storage temperatures and 3,000 at freezing temperatures. These figures do not include the refrigeration required to cool the goods.

1545. Refrigeration Required to Cool the Articles.—The amount of refrigeration required to reduce the goods from their temperature t to the temperature t_1 of the room is given by the formulas

$$H = (w_1 s_1 + w_2 s_2 + w_3 s_3 + \ldots)(t - t_1), \qquad \textbf{(139.)}$$

$$F = \frac{H}{385{,}300} =$$

$$.000003505\,(w_1 s_1 + w_2 s_2 + w_3 s_3 + \ldots)(t - t_1), \qquad \textbf{(140.)}$$

in which w_1, w_2, etc., are the weights in pounds of the different kinds of produce to be cooled, s_1, s_2, etc., their corresponding specific heats, and H and F are refrigeration units in B. T. U. and tons of refrigeration, respectively.

1546. The following table contains the specific heats of several articles, together with their latent heats of freezing:

TABLE 34.

SPECIFIC HEAT AND LATENT HEAT OF FREEZING OF VICTUALS.

Substance.	Specific Heat.	Latent Heat of Freezing, B. T. U.
Beef	.80	110
Veal	.70	90
Mutton	.80	110
Pork	.60	72
Eggs	75	100
Vegetables	.90	125
Cream	.70	90
Milk	.90	125
Fish	.85	115
Lobster	.80	100
Oysters	.85	115
Chicken	.80	110

EXAMPLE.—Find the refrigeration required for a room 35′ × 50′ × 12′ in which are placed 25,000 pounds of beef daily, the temperature of the meat being 95°, that of the atmosphere 70°, and that of the rooms 35°.

SOLUTION.—(1) The refrigeration required to keep the room at 35° is given by formula **137.** Here we have

$$\begin{aligned}\text{Area of walls} &= 2 \times 35 \times 12 + 2 \times 50 \times 12 = 2{,}040 \text{ sq. ft.}\\ \text{Area of floor and ceiling} &= 2 \times 35 \times 50 = \underline{3{,}500 \text{ sq. ft.}}\\ A &= 5{,}540 \text{ sq. ft.}\end{aligned}$$

The values of t and t_1 are 70° and 35°, respectively, and therefore $t - t_1 = 35$. Taking the value of $c = 3$, the formula gives

$$F = .000003505 \times 3 \times 5{,}540 \times 35 = 2.04 \text{ tons.}$$

(2) The amount of refrigeration required to cool the beef to the temperature of the room is given by formula **140.** Here $t = 95°$, $t_1 = 35°$, $t - t_1 = 60$, $w_1 = 25{,}000$, $s_1 = .8$ (see Table 34), and since there is no other article, the terms $w_2\,s_2$, $w_3\,s_3$, etc., are left out; hence,

$$F = .000003505 \times 25{,}000 \times .8 \times 60 = 4.21 \text{ tons.}$$

The total amount of refrigeration required is, therefore,

$$2.04 + 4.21 = 6.25 \text{ tons.} \quad \text{Ans.}$$

METHODS OF COOLING STORAGE ROOMS.

1547. General Considerations — The Three Methods of Cooling Generally Used. — The first requirement of a cold-storage room is good insulation. The best methods and materials for insulation have already been described. All windows and other openings by which light is admitted to the room should be closed with light-tight shutters, so as to prevent any daylight from entering. If this is not done, radiant heat will enter the room. Such windows can be closed with large shutters on the inside, similar to ice-house doors, which can be opened for ventilation when desired. A cold-storage room should be provided with some means of ventilation, so that the rooms can be thoroughly sweetened and the air renovated between seasons when the rooms are not in use. Modern houses are equipped with a ventilating device, to be presently described, which permits of the ventilation of a cold-storage room while in use.

There are three methods employed at present for maintaining the temperatures of cold-storage and freezing rooms. The first is by means of direct radiation, where the brine pipes are run in the room and the brine is allowed to circulate through them. This was the original system and is the simplest and cheapest one to install. The second is by indirect radiation, where the pipe coils are place in a coil bunker on the upper floor and the air is allowed to fall to the floors below by gravity, and on being warmed is returned to the coil bunker by means of flues or ducts arranged for that purpose. The third and most approved method of cooling is by means of a fan or blower, which sets the air in circulation. This system is quite similar to the blower system of heating and ventilating. It consists of a fan or blower, which draws the air from the cold-storage rooms, blows it through a system of pipe coils, and then delivers the cold air back to the cold-storage room by means of ducts arranged specially for the purpose.

1548. Direct Radiation.—In this method of cooling, the most important feature is the arrangement of the pipe coils. These coils should be kept away from the wall at

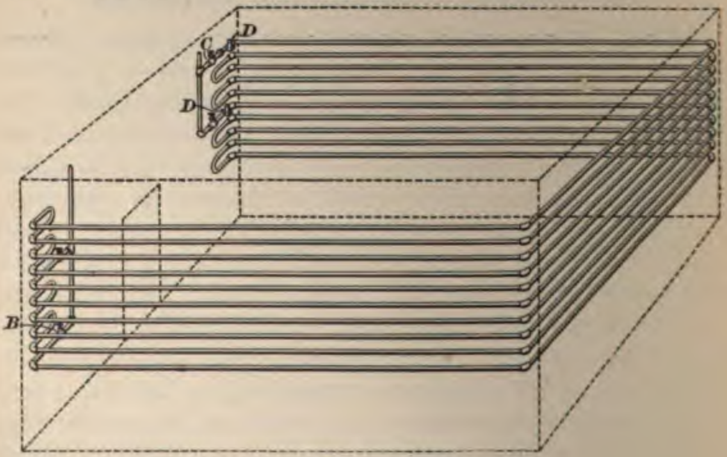

Fig. 355.

least 6 inches by means of furring studs nailed fast, on which the pipe hangers carrying the coils are fastened. This allows a circulation of air between the coils and the wall and prevents the frost from reaching the insulation in case of a heavy coating. There should be at least two coils to a room, and if possible more, so that the temperature can readily be regulated by means of cocks or valves provided on the inlets and outlets of these coils. The usual method is to allow the brine to enter the bottom of the coil and come out of the top, the outlet near the top being provided with an air-cock for the removal of air.

Fig. 355 shows a cold-storage room with the door at one end and provided with two coils of the return-bend pattern. The inlet to the coils is shown at *B*. The outlet is provided with a cock *C* and an air-valve *D*. These are both quite near the door, so that if the room is well filled with goods, the floors can be readily cut out for the purpose of

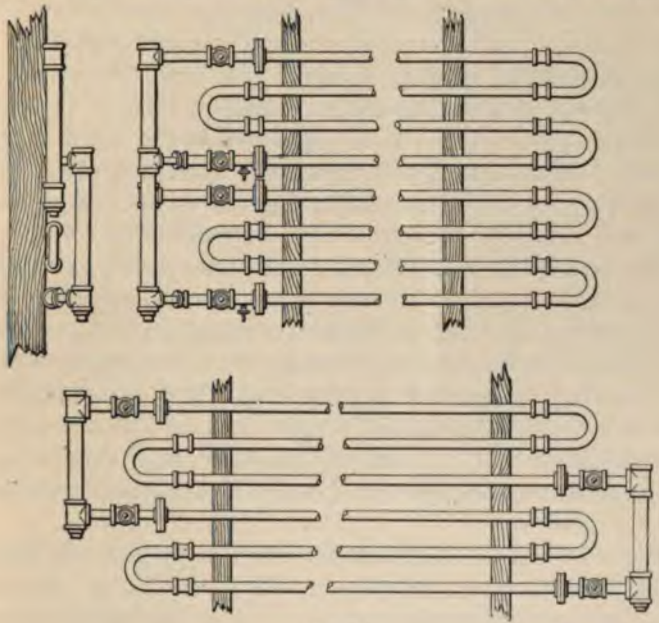

FIG. 356.

regulating the temperatures and also closing off the coils in case of a leak. Fig. 356 shows several methods of coil connections in detail. The upper coil should be as close to the ceiling as possible, so as to prevent any chances of a warm-air pocket lying next to the ceiling. In case a girder passes through the room, it is best to elbow the first and second pipes under it, so as to thoroughly cool the upper portion of the room. The arrangement of the coils should be such that all portions of the room will be equally cooled by each coil. In very large rooms this is impossible, but ordinarily the coils run around all the walls of the room so as to permit this. Drip-pans should be provided under the coils for catching the melted frost when the coils are shut off. The latter should drain into a tub or barrel at one end. It is best not to connect these drip-pans into a series of drain-pipes, as these connections are liable to freeze up in being carried through freezing rooms; and even if kept clear, they let in from outside a certain amount of warm air laden with moisture. The temperature in the room is regulated by either closing off or partially throttling the outlets to the various coils. In throttling the outlet, the coil is kept full of brine, and there is no opportunity for air to accumulate, as there is in case of the inlet being throttled. 1¼-inch full-weight black iron pipe is commonly used for brine circulations. If the brine temperature is 0°, 1 lineal foot of pipe per 15 cubic feet will give a temperature of 30°; 1 lineal foot of pipe per 6 cubic feet will give a temperature of 10°, and 1 lineal foot of pipe per 3 cubic feet will give a temperature of 7°. With 15° brine, 1 lineal foot of pipe per 10 cubic feet will give 30°, and 1 lineal foot per 4 cubic feet will give 20°.

As a rule, all freezing rooms are equipped with direct piping, as there is no objection to the goods being near the pipes, and an exact temperature is not necessary, two or three degrees variation having no effect on the quality of the goods. For ordinary purposes, such as the keeping of tierced meats, butter, etc., direct-piped rooms may be used; but for good results in keeping fruit, cheese, eggs, etc., the indirect method or forced-air circulation is recommended. In case a house is already piped with direct radiation, this can be modified so that it will give the same results as indirect radiation, by putting aprons over the coils, as shown in Fig. 357, which represents a section of the room, on the walls of which are placed 2″ × 6″ planks *A* carrying the coils of the pipe *B*. These are provided with a drip-pan *C*. In front of these coils, a number of 2″ × 4″ studs *E* are fastened, and to these are nailed the boards *D*; the studs are toenailed to the ceiling. A false ceiling *F*, slanted slightly towards the coils, is built a few inches below the ceiling of the room. This arrangement creates a natural circulation or draft, as shown by the arrows. The warm air passing to the top of the room is cooled by means of the coils, falls back of the aprons *D*, and goes out of the opening at the bottom. With this arrangement and proper temperatures, eggs may be carried with results almost as good as those of a forced-circulation house, and better than those of an indirect-radiation house, the only disadvantage being that the coils are in the same room; but if these are carefully attended to and the drip-pans are kept clean and empty, the results will be as satisfactory as those obtained by the indirect method.

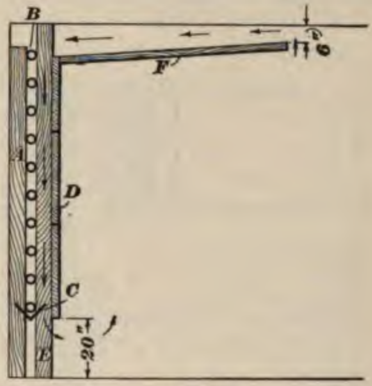

FIG. 357.

1549. Indirect Radiation.—Fig. 358 shows the section and Fig. 359 a plan of a cold-storage warehouse arranged for indirect radiation. It consists of the coil-bunker room *A*, in which are placed the coils *B*, on posts. These are provided with a drip-pan *C*. The cold-air duct *D* communicates with the room *E*. Alongside of *D* is another cold-air duct *D′* communicating with the room *G*, and a third *D″*

communicating with the room *F*. (See Fig. 359.) These ducts are provided with dampers *H*, *H*. The warm air from the room *E* passes up the duct *K*, and so to the top of the room *J* into the coil bunker *A*, where it is cooled and again circulates through the various rooms. The path of the air is shown by arrows.

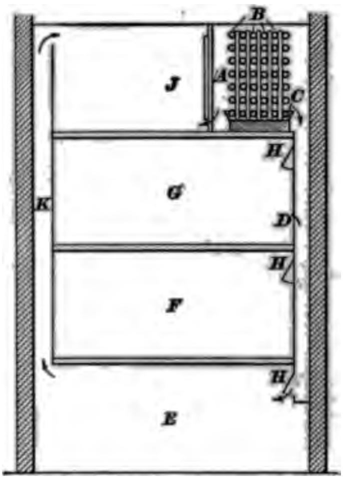

FIG. 358.

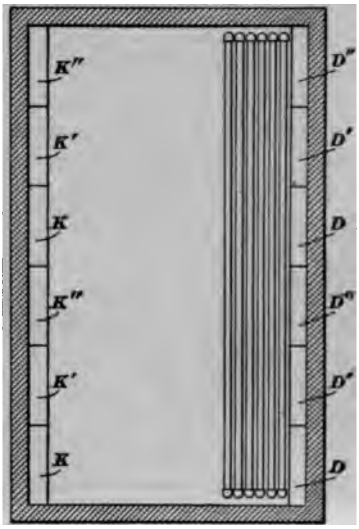

FIG. 359.

By this arrangement, any of the rooms may be closed off by shutting the dampers *H*. The duct *K* should be 25 per cent. larger than the duct *D*; and the duct *D* should have an area of 3 square inches per square foot of floor area of the room. This applies to cold-storage rooms having a clear ceiling height of about 10 feet. This proportion between the area of the duct and that of the room will give a good circulation of air for a difference of temperature of 2° between the floor of the room *E* and the ceiling of the room *J*.

Fig. 359 gives a plan of the coil bunker and the room *J*, showing the subdivision of the ducts for the various lower rooms, as D, D', D'', and K, K', K''. By having all the coils in one coil bunker, as shown here, the advantage is gained that the temperature may be ascertained without going in more than one room, and therefore without letting heat into the other rooms by opening them. In this way it is comparatively easy to keep a constant temperature. Besides, the complete turning off of one coil will only have one-quarter of the effect on the temperature that it would have if each room had its independent series of coils, for if each room had two coils, the shutting off of one would cut down the radiating surface in the room 50 per cent. If, on the other hand, the coil bunker is supplied with 8 coils, the shutting off of any one of them cuts down the radiating surface but 12½ per cent. Furthermore, all the drip from the coils is concentrated in the coil bunker, and when the frost is scraped off the pipes, it falls into the drip-pan *C* and can easily be removed.

The disadvantage of this system is that all the rooms should be devoted to the carrying of one product; consequently, if there are not enough goods to fill all the rooms, some space is lost, and no other goods can be placed in the vacant rooms. This is specially true in the case of eggs, for which this system is particularly adapted. The quantity of pipe required to maintain a certain temperature is essentially the same as that required with direct radiation.

1550. Forced-Air Circulation.—The general arrangement of an air-circulating system is shown in Fig. 360. The circulation is accomplished by means of a fan or blower *B*, which is placed in the cooling room *A*. The air is drawn from the various cold-storage rooms through the duct *E* into the central opening in the fan case, and is driven by the

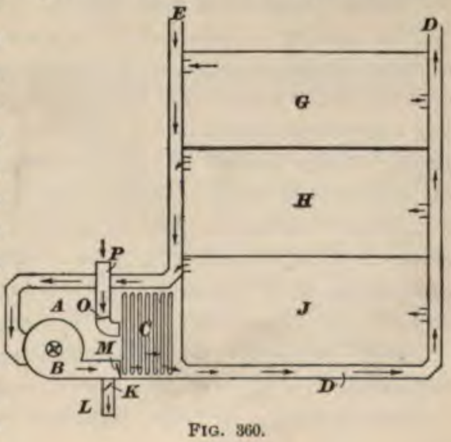

FIG. 360.

action of the fan through the brine coils *C* and out through the delivery duct *D* into the rooms *G*, *H*, and *J*.

An advantage of this system is that the bunker room *A* can be placed on the first floor or in the basement of the building, as it is not necessary to have this room either directly above or below the various cold-storage rooms. The drip from the coils of the bunker will in no way injure the goods stored in the rooms, as in the case of the indirect-radiating system, when the catch pan leaks.

Another advantage of this system is that the rooms may be easily ventilated. This is done in the following manner: The fan *B* is provided with an extra outlet pipe *L*, in which is a damper *K*. When the fan is running, the air is drawn from the various rooms, and by opening the damper *K* and closing the damper *M*, the air, instead of passing through the coils *C*, is blown outdoors through the pipe *L*. The damper *O* in the pipe *P* is then slightly opened; this permits fresh air to enter the coil bunker *A*. The air passes through the coils *C* and is cooled by means of them to the required temperature after it enters the rooms *G*, *H*, and *J*. This arrangement allows all the dead air to be drawn from these various rooms and fresh air to be admitted. In some houses, a separate series of coils and ducts, entirely independent of the main circulating system, are arranged for the purpose of ventilating the various rooms.

This indirect forced-draft system is especially good for storing eggs; it creates a circulation in the rooms, and consequently keeps the air thoroughly fresh, besides keeping the temperature very nearly uniform throughout the room. Care should be taken, however, not to run the fan or blower too fast, as too much air circulation tends to evaporate the eggs. The best way to regulate the amount of air is to run the fan at such a speed as to keep a difference of about 2° between the temperature of the inlet air to the room and that of the air returning to the bunker. This difference should not be less than 1°, or a good deal of evaporation will be noticed in the eggs.

The quantity of pipe required with a forced draft is about the same as for direct radiation, or, approximately, 1 foot of pipe for every 30 cubic feet of space. The coils of pipe should be so arranged in the bunker as to break the currents of air and yet be far enough apart so that there is no danger of a solid wall of snow forming between the coils.

1551. Sharp Freezing.—For the purpose of freezing fish rapidly, a **Sharp Freezer,** such as is shown in Fig. 361, is employed. This consists of a series of return-bend coils, one back of the other, the pipes of which form a series of shelves, one under the other. The brine or direct-expansion ammonia is allowed to enter the top header connecting these various coils, and the return is taken from the bottom header. On the shelves formed by the pipes are placed galvanized-iron pans, 20″ × 36″ at the bottom and about 4 inches deep, filled with fish The doors covering these coils are then closed and the fish are allowed to remain in the freezer for 15 to 20 hours. They are then

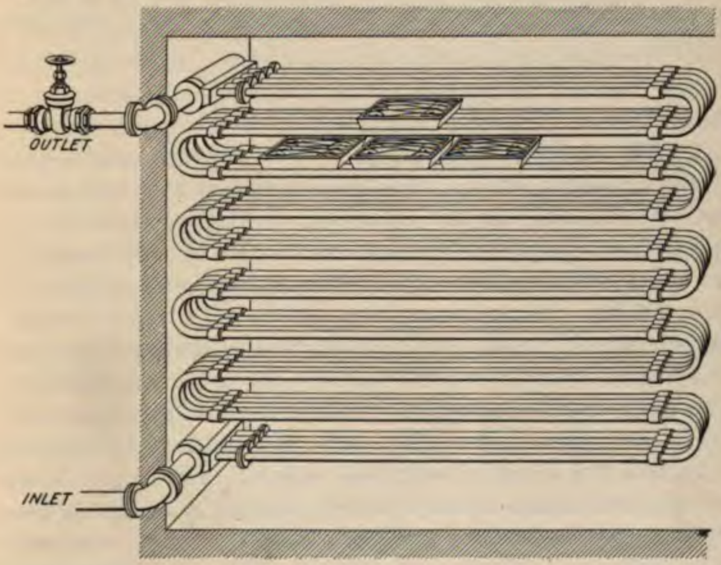

FIG. 361.

removed from the pans and packed in boxes about the size of the pan. Before the fish are placed in the pans, they are usually dipped in cold water, so as to remove any salt water or fish oil that may adhere to them; this also forms a thin coat of ice, which helps to preserve the fish. If brine circulation is used, a temperature of 10° to 15° below zero is required to obtain a temperature of 0° in the freezer.

If a plant is insulated for the sole purpose of freezing fish or meat, an allowance should be made at the rate of one ton of refrigerating effect per ton of meat or fish frozen in every twenty-four hours, in addition to the space cooled.

STORING VICTUALS.

1552. Storage of Eggs.—For the successful cold storage of eggs, a room should be equipped with either the indirect-radiating or the forced-air-circulating system. If this is not possible and direct radiation is already installed, the coils should be provided with aprons, as explained in Art. **1548.** A temperature of 30° to 31° should be maintained in the rooms, and the relative humidity of the air should be kept at between 65 and 70 per cent. of saturation. The eggs should be placed in new cases which have been thoroughly dried, together with their fillers, by placing them in a dry house or over the boilers. The eggs should also be thoroughly candled before being put in the house, and only those that are perfectly sound should be stored. When the eggs are taken out, they should be placed in a comparatively cold room, so that they will get warm gradually, which will prevent them from sweating to any great extent.

1553. Storage of Dairy Products.—Milk and cream can be kept for a considerable length of time if properly cooled before storing. This cooling is effected by means of a cooler built on the same general plan as the Baudelot cooler used in breweries, but having tinned-copper pipes instead of iron pipes for the circulation of brine.

When the milk is allowed to run over these pipes, all the animal odor is eliminated, and the milk preserves its sweetness much longer than when placed in cans while warm and then put in a cold-storage room.

Cheese is usually kept at a temperature of 34° to 36°. The room should be kept comparatively dry, with a relative humidity not exceeding 70 per cent., so as to avoid molding. A temperature lower than 34° is likely to make the cheese granular, and does not help to preserve the quality any better than the higher temperature.

Butter is now frozen and kept at a temperature of 10°. This temperature has been found to preserve the flavor much better than the higher temperatures used in former days. Butter is usually stored in 60-pound firkins. Small packages are not carried successfully, as it is very difficult to prevent the outer crust of the butter from losing its flavor and becoming somewhat stale. It is therefore better to use large firkins and make prints after the butter is taken from the storage rooms than to carry the prints themselves in storage.

1554. Storage of Fruit.—Winter apples are carried at temperatures as low as 28°, which is just above their freezing point. Care should be taken that the fruit is of a good quality and perfectly sound, in order to prevent decay, which causes fermentation, with the production of carbonic-acid gas and a disengagement of heat. This heat will raise the temperature of the room, thereby aiding in the decomposition of other apples in the immediate neighborhood. The carbonic-acid gas resulting from fermentation is also rather detrimental to the proper keeping of fruit.

Bartlett pears should be picked green and stored in boxes or small barrels. They can be carried successfully from 60 to 90 days, if care is taken to keep the room dark and at a temperature of 32° to 34°.

All citrus-fruits, such as lemons and oranges, should be carried at a temperature of 33° or 34°, and should be entirely isolated from any dairy products and eggs. If possible, rooms should be arranged that have separate entrances from those going into the egg and butter rooms. The odor from lemons and oranges is very penetrating and gives a very disagreeable flavor to eggs, making them almost unsalable.

1555. Temperature Required by Different Articles.—Table 35 gives the temperatures at which several goods are usually carried. The usual rule is to carry an article just above its freezing point. There are a few exceptions to this rule, such as cheese; but if care is taken to carry the room just above the freezing point and keep the air sweet and dry, good results are almost always attained.

TABLE 35.

STORAGE TEMPERATURES OF VARIOUS PRODUCTS.

Article.	Storage Temperature.
Apples	30°
Berries	34°
Butter	10°
Beer	36°
Cheese	34°
Dried Fruit	36°
Eggs	30°
Fresh Meat	33°
Frozen Meat	20°
Fish	15°
Furs	5°
Grapes	33°
Lemons	33°
Oranges	33°
Peaches	34°
Pears	33°

REFRIGERATION FOR PACKING HOUSES.

1556. Chilling, Storage, and Freezing Rooms.—In packing houses there are two kinds of rooms to be cooled, viz., **chilling rooms** and **storage rooms.**

Chilling rooms are those in which the carcasses are placed immediately after the animals have been slaughtered, in order to cool them from 95° (blood heat) to the storage temperature, which is about 35°. The temperature of chilling rooms is kept at about 28°.

After the meat has cooled to about 35° or 32°, it is taken to the storage rooms, where it is kept. The temperature of these rooms is about 35°.

It is often desirable to freeze the meat. This is done in special rooms called **freezing rooms,** which are kept at a temperature of 10° or under.

1557. Refrigeration Required.—To determine the exact amount of refrigeration required, the chill rooms, cellars, storage rooms, etc., should be carefully measured; the amount of refrigeration is then found by the rules given under cold-storage refrigeration. To this should be added the refrigerating effect required to cool the meat, as given by formula **140.** Beef, after being slaughtered, is usually hung in a well-ventilated room for several hours. The temperature of the meat, however, is reduced but little, and so it is customary to figure on the supposition that the beef enters the chill room at a temperature of about 95° (this is the value of t in formula **140**). The beef must then be cooled to at least 35° (this is the value of t_1 in the same formula). The same applies to other animals.

If the weights of carcasses of different kinds are represented by w_1, w_2, w_3, etc., and the number of carcasses of each class by n_1, n_2, n_3, etc., the weights to be cooled will be $n_1 w_1$, $n_2 w_2$, $n_3 w_3$, etc. Substituting these values for w_1, w_2, etc., in formula **140,** and putting $t - t_1 = 95° - 35° = 60°$, we get, for the tons of refrigeration required,

$$F = .000003505\,(n_1 w_1 s_1 + n_2 w_2 s_2 + \ldots\ldots) \times 60$$
$$= .0002103\,(n_1 w_1 s_1 + n_2 w_2 s_2 + \ldots\ldots). \qquad \textbf{(141.)}$$

EXAMPLE.—In a certain abbatoir there are 5,000 cubic feet of chill-room space, 10,000 of pickling space, and 20,000 of storage. 100 hogs and 30 beeves are killed daily, the average weight of the hogs being 240 pounds and that of the beeves 750 pounds. How much refrigeration is required?

SOLUTION.—Total space = 5,000 + 10,000 + 20,000 = 35,000 cu. ft. According to Art. **1544,** 1 ton of refrigeration will be required for 6,000 cu. ft. of space. Therefore, the refrigeration required to cool 35,000 cubic feet is

$$\frac{35{,}000}{6{,}000} = 5.83 \text{ tons.}$$

To apply formula **141,** we have $n_1 = 100$, $n_2 = 30$, $w_1 = 240$, $w_2 = 750$, $s_1 = .6$, $s_2 = .8$ (see Table 34). Substituting these values in the formula, we get

$$F = .0002103\,(100 \times 240 \times .6 + 30 \times 750 \times .8) = 6.81 \text{ tons.}$$

The total amount of refrigeration required is, then,

$$5.83 + 6.81 = 12.6 \text{ tons, nearly. Ans.}$$

1558. For rough calculations, 1 ton of refrigeration may be taken to cool about 4,000 cubic feet of chill-room space and 8,000 to 10,000 cubic feet of storage or pickling room space.

In chilling, 1 ton of refrigeration will suffice for

20 hogs (average weight 250 pounds),
or 7 beeves (average weight 700 pounds),
or 50 calves (average weight 90 pounds),
or 70 sheep (average weight 75 pounds).

For freezing rooms, 1 ton of refrigeration per ton of meat frozen, including the space in which the meat is carried, is a good allowance.

1559. Chill Room.—The indirect system should be used in piping chill rooms for beeves, hogs, sheep, etc. If there is sufficient headroom, the pipes can be placed overhead in a bunker formed by means of drip-pans, as shown in Fig. 362, the beams for carrying the meat track forming the supports for the pans. If, however, the chill rooms were originally on the ice-bunker system, having the ice chamber overhead, the latter can be piped, and the drip from the coils will then be caught by the ice pans. By this arrangement, the steam which passes off from the meat is condensed in the ice chamber or coil bunker overhead, and to a large extent takes away the animal odor. Chill rooms piped on the sides do not accomplish this, and often the meat coming from them is sour.

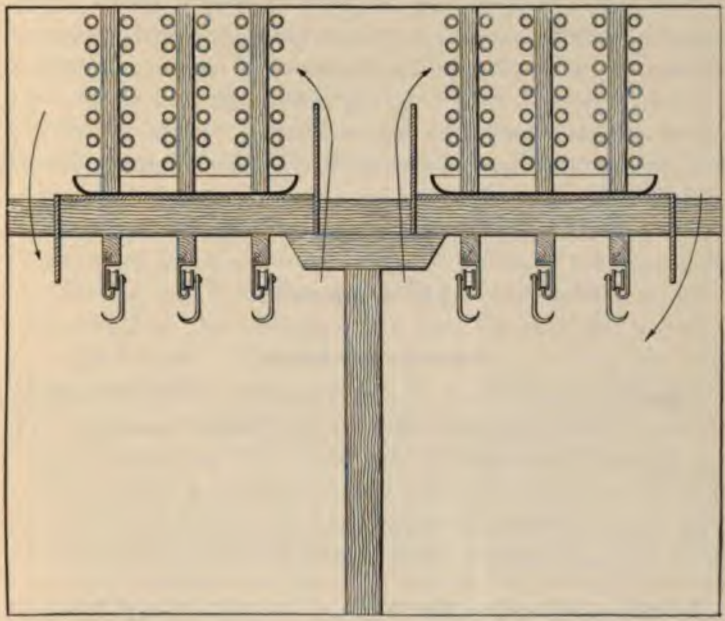

FIG. 362.

The quantity of pipe required to cool chill rooms is about as follows: For direct expansion, 1 foot of 2-inch pipe will suffice for 13 cubic feet of chill-room space, and 1 foot of $1\frac{1}{4}$-inch pipe will suffice for 9 cubic feet. In case of brine circulation with 15-degree brine going to the chill room, 1 foot of $1\frac{1}{4}$-inch pipe will suffice for 5 cubic feet of space. If the brine is at 0° or colder, the amount of pipe required will be the same as that given for direct expansion.

1560. Storage Rooms.—Storage or pickling rooms are piped overhead, if there is sufficient headroom; otherwise, on the side walls or along the posts. Their temperature is seldom below 5°, as pickling is prevented if the temperature goes below that point. These rooms are often kept at temperatures as high as 42° and 45°.

The quantity of piping required in case of direct expansion is as follows: 1 foot of 2-inch pipe will suffice for about 40 cubic feet of well-insulated space; 1 foot of 1¼-inch pipe will suffice for 30 cubic feet of space. For brine circulation with 15-degree brine, 1 foot of 2-inch pipe will cool 15 cubic feet of space, and 1 foot of 1¼-inch pipe will cool 10 cubic feet of space.

ICE MAKING.

SYSTEMS USED.

1561. There are now two systems in use for making ice, viz., the **can system** and the **plate system.** The can system is the more common of the two, being cheaper in first cost and requiring less attention in manipulation. The plate system, however, has the advantage of being more economical in the end and of giving a clearer ice.

1562. The Can System.—The apparatus used in the can system consists of a large rectangular wood or iron tank containing the expansion coils or pipes. Galvanized-iron cans are placed between the rows of expansion coils. These cans are filled with distilled water, and when the brine is chilled below the freezing point, the water in the cans freezes. If the temperature of the brine is not allowed to fall below 25° and ordinary well-water is used in the cans, the ice produced will be comparatively clear on the outside and rather snowy in the center. If, however, the brine temperature is allowed to fall to about 15°, the ice will be entirely opaque.

To get good, clear ice, distilled water is used. This makes a bright, clear ice, with the exception of a small core or feather in the center. It is necessary, however, to have a good distilling apparatus. The white appearance in ice made from non-distilled water is due to minute air-bubbles which are held in suspension in the water and frozen in the ice, forming a sort of snow. In case of distilled water, this air is eliminated by boiling and subsequent evaporation. From this it will be seen that it is necessary to make enough distilled water in every twenty-four hours for the nominal ice-making capacity of the plant.

In the ordinary ice factory, the distilled water is usually made by condensing the exhaust steam of the engine operating the compressor in case of a compression plant, or by cooling the condensed steam that leaves the generator, still, or retort of the absorption plant. Ordinarily the steam that is required by a first-class machine of either kind is considerably less than the amount of ice that the machine can make in a given time. It is therefore necessary to draw live steam from the boiler and condense it, in order to make up the deficiency. It follows from this that the economy of the plant depends entirely upon the economy of the boiler.

The ordinary yield of a can plant is about 6 pounds of ice per pound of coal. There is a certain loss due to exhaust thrown away from auxiliaries, steam leaks, etc., that brings the boiler evaporation down to this figure, though there are some tests on record where a plant has done as well as 8 pounds of ice per pound of coal. The ordinary water consumption for condensing purposes, boiler, etc., is 4 gallons per minute per ton of ice made in 24 hours, the water being well-water at 60° or colder. With warmer water, a larger quantity has to be used.

1563. Plate System.—In the plate system, the refrigerating fluid is circulated through vertical cast or wrought iron hollow plates set on edge in a tank of water. These plates are usually about 16 feet long and 8 feet high. Ice begins to form on the plate and gradually extends out into the tank. If the temperature of the plate is kept comparatively high at the start, until 2 or 3 inches of ice is formed, and then gradually reduced as the ice formation increases, a clear cake or plate of ice will be formed on each

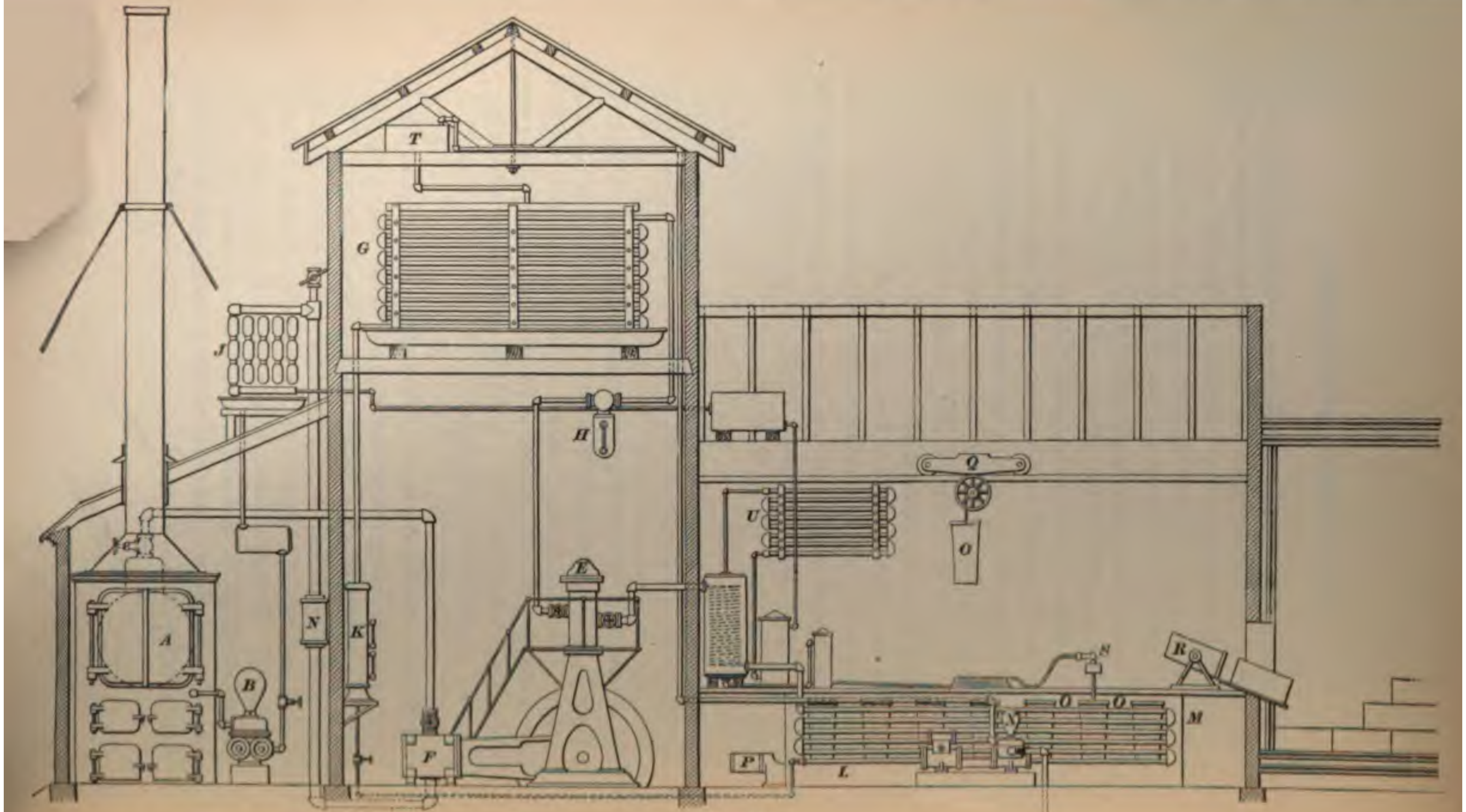

FIG. 303.

side of the plate. The refrigerating fluid is then drained from the plate, and warm water is introduced, which thaws the ice adhering to the sides. As soon as the ice is detached, it floats to the surface of the water in the tank, and is then drawn up by means of a large traveling crane and cut up on a special table fitted for the purpose into cakes of the proper size.

From the above description, it will be seen that ordinary filtered water can be used in the cells of the plate system, and the expense of distilled water is avoided. Great care, however, must be exercised to prevent the plate from freezing too rapidly at the start, in order to avoid the formation of white ice.

Plate ice, in the average ice-making plant, requires from 8 to 12 days to freeze, according to the thickness of the cake and the quality of the water from which the ice is made. In the can system, 24 to 60 hours is all the time required to freeze a cake of ice, according to the size of the cake. The difference in time in the two systems is due to the fact that in the plate system, freezing proceeds from one side only, whereas in the can system it proceeds from the two sides and the end.

The first cost of a plate plant is greatly in excess of that of a can plant. The floor area is also greater for the plate plant, but a good plant of this kind, properly put up and operated, is, once installed, by far more economical than a can plant, and gives much better ice.

COMPRESSION ICE-MAKING PLANT WITH CAN SYSTEM.

1564. Division of the Plant.—A complete can plant consists of the following parts:

(*a*) Steam-boiler plant.
(*b*) Refrigerating or ice-making machine.
(*c*) Freezing tank with accessories.
(*d*) Distilled-water system.

A complete can-system compression plant is shown in Fig. 363.

1565. Steam Plant.—For the purpose of getting good, clear can ice, it is necessary to have a steam-boiler of ample capacity, so that there will be no danger of the boiler foaming. A boiler that does not give good, dry steam invariably causes trouble in an ice plant, the ice becoming discolored and cloudy on account of the priming. Five horsepower per ton of ice to be made is a good allowance for the steam-boiler plant.

A well-equipped ice-factory boiler plant consists of the following parts (see Fig. 363):

(*a*) A good boiler with fixtures and stack.

(*b*) A boiler-feed pump.

(*c*) An injector, to be used in case of accident to the feed-pump.

(*d*) A feed-water heater.

In countries where the water is bad and contains alkalies, the steam-boiler plant should be in duplicate, and in case the water contains sulphates, a live-steam purifier can be used with advantage. Under these conditions, it is best to change boilers once every week or two, giving the spare boiler a thorough cleaning, removing all scale, etc.

1566. Refrigerating-Machine (see Fig. 363).—This consists of the ammonia compressor *E* with engine *F*, ammonia condenser *G*, oil trap *H*, receiver *K*, and expansion coils *L* in the ice tank *M*. Usually the ice-machine is placed in a room by itself, to prevent the dust from the boiler affecting the refrigerating machinery. The compressor is placed upon a solid stone foundation, and the condenser, usually of the atmospheric type, is placed above the compressor in a latticework tower, sufficient height being given to the condenser so that the water passing from it will flow to the exhaust-steam condenser *J*, and from there to the boiler-feed pump *B*. This allows the water to get quite warm before it is sent through the feed-water heater to the boiler.

To get economical results with a compression machine, the engine should preferably be equipped with some form of releasing valve-gear, as the Corliss. A plain slide-valve engine will use more steam to operate the compressor than the compressor would be able to freeze into ice. The common practice is to allow 2 tons of refrigerating effect per ton of ice to be made. The exhaust from the engine *F* is piped into the exhaust-steam condenser *J*, as shown in the figure. It is from this exhaust steam that the ice is made after the grease has been extracted by means of a separator *N*. If the engine requires more steam to operate it than it can freeze into ice, the machine is not economical, as a certain amount of exhaust steam has to be allowed to go to waste. With the usual allowance of 2 tons of refrigerating effect per ton of ice, and assuming $1\frac{1}{2}$ horsepower in the steam engine to be required per ton of refrigeration, the power required will be 3 horsepower per ton of ice. This gives one-third of a ton, or about 667 pounds of steam per horsepower in 24 hours, or 28 pounds of steam per horsepower per hour. As an ordinary slide-valve engine uses from 40 to 50 pounds of steam per horsepower per hour, it appears that an engine with a releasing gear or some form of expansion-valve must be used, if the best effect is desired.

1567. Ice Plant.—The ice-making plant comprises the following parts (Fig. 363):

(*a*) Ice-making tank *M*.
(*b*) Ice cans or molds *O*, *O*.
(*c*) Grating for holding cans in position.
(*d*) Ice-can covers.
(*e*) Insulation of bottom and sides of tank.
(*f*) Brine agitator *P*.
(*g*) Crane *Q* with geared hoist and can lift.
(*h*) Ice-can dump *R*.
(*i*) Can filler *S* with hose.
(*j*) Expansion coils *L* with headers and valves.
(*k*) Brine hydrometer and thermometer.

1568. Ice tanks are usually made of steel, though in some plants wood is used. Steel tanks should be made of

¼-inch steel for tanks 3 feet or more in depth, $\frac{3}{16}$-inch steel being heavy enough for tanks 30 inches deep. The tank should be properly braced and reinforced and have an angle-iron rim punched for bolt-holes around the top, so that the grating can be securely attached thereto.

1569. Expansion coils should be of extra heavy pipe and should run the full length of the tank, one coil between each row of cans. The coils should be continuously welded throughout their lengths, with tails carried through the end of the tank, stuffing-boxes being provided on the tank to prevent brine leakage. The coils are usually strapped in such a manner as to permit the grating to be supported by the straps, as shown in Fig. 364. There should be 100 square feet of coil surface for each ton of ice made. Each coil should be provided on its inlet with an expansion-valve and the outlet should be provided with a stop-valve.

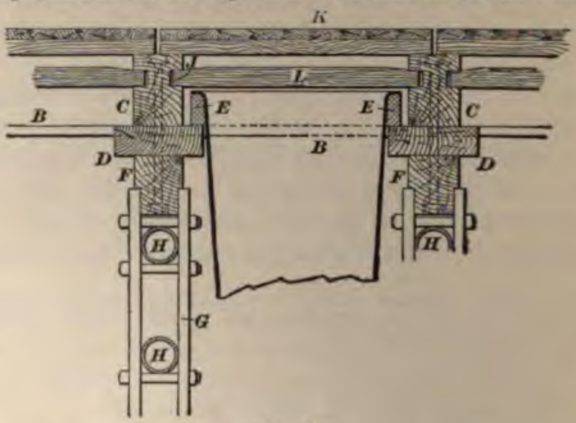

FIG. 364.

1570. Ice Cans.—The ice cans or molds in which the ice is frozen are usually made of various sizes, and the cakes of ice formed in them usually weigh between 50 and 300 pounds.

To find the net weight in pounds of the cake formed in a can, find the volume of the can in cubic feet; multiply the result by 57.2 (weight of a cubic foot of ice) and take 5 per cent. off the product (this to allow for thawing). The result will be the required net weight.

The can should generally be made of 16-pound galvanized iron, although for small ones 18-pound galvanized iron may be used. The best cans are made of what is known as "Best Bloom stretched and patent leveled galvanized iron." Such iron is perfectly flat, and has no inequalities in it. Ice cans are usually made with a slight taper from the bottom upwards, which permits the ice to slide out easily after it has been thawed free of the can. The smaller sizes of cans have one joint along the sides, near one of the corners, and the larger sizes one or two joints. These joints should be riveted on 1-inch centers, with all rivet heads driven close. The seams should then be soaked with solder and floated flush. The bottoms are usually flanged and inverted into the body, the joint being made by riveting and soldering in place. The top of the can should be reinforced with a band of flat iron, and the edge of the can should be wired over. Two holes are punched in the long side of the can, under the band, for lifting.

1571. Tank Surface.—With brine at 15° in the ice tank, and a good circulation, an 8-inch ice can will freeze solid in about 24 hours, an 11-inch can in 30 to 36 hours, and a 14-inch can in 40 to 48 hours. In very warm weather, the brine sometimes rises to a temperature higher than 15°. Ice plants are therefore usually constructed with a real capacity greater than their nominal capacity; the ratio of the former capacity to the latter is called the **surface** of the tank. For example, a 10-ton plant having 2½ surfaces would indicate that the total capacity of the ice cans was $2\frac{1}{2} \times 10 = 25$ tons. Plants having 8-inch cans are constructed with surfaces of 1½ to 2; those having 11-inch cans with surfaces of 2 to 2½, and those having 14-inch cans with surfaces of 2½ to 3. These proportions permit of a slight rise in the brine temperature and also allow a certain quantity of ice to be carried frozen in the cans, which can be pulled in case of great demand.

1572. Grating and Covers.—The ice-can grating is usually made of hard wood, such as oak or ash, firmly bolted together with galvanized-iron cross-straps. Such a grating is shown in Fig. 364. It will be seen that the rim *E* of the ice cans rests on the galvanized-iron cross-strap *B*, which is mortised into the oak strip *D*. The upper strip *C* on which the can covers *K* rest is mortised out with a groove *J*. In this is placed a stick or button *L*, which firmly buttons the can down, so that it can not float up. The oak strip *F* is supported by means of the expansion-coil straps *G*, which are mortised into it, and which in turn support the expansion coil *H*. Through bolts firmly bolt *C*, *D*, *F*, and *B* together, making a very rigid framework. The can covers are usually made of two pieces of oak, well nailed together, with two layers of good insulating paper in between. The ends of these covers are sometimes hollowed out with a recess, which permits the cover to be raised easily, while other covers are provided with regular plates and handles for lifting.

1573. Insulation.—The ice tank should be insulated in the manner described in Art. **1479**, the grating being allowed to project over the edge of the tank and the insulation being nailed fast thereto.

1574. Agitator.—In order to get a high efficiency from the expansion coils in the tank and freeze the ice rapidly, agitators are used. They are of three classes, viz., centrifugal pumps, displacement pumps discharging into distributing pipes, and propellers.

The object of the centrifugal pump is to circulate a large quantity of brine, and thereby keep a comparatively even temperature. The usual method is to take the brine from one corner of the tank and discharge it into a header along the opposite side. This insures a comparatively rapid circulation and causes the ice to form much faster in the ice cans.

When, however, it is desired to have some cold storage in connection with the ice plant, and it is necessary to use a displacement pump, such as an ordinary steam-pump, to circulate the brine throughout the cold-storage rooms, the centrifugal pump can be done away with by properly arranging distributing pipes in the brine tank. Such arrangement is shown in Fig. 365. The brine-pump *B*, drawing the brine from near the top of the ice tank *A*, discharges it through the coils in the cold-storage room *C*. After passing through these coils, it returns to the brine tank and enters the header *D*; from there it is distributed through the distributing pipes *F*, which are perforated with small holes. It is advisable to place one of these distributing pipes under each line of expansion coils, as this permits the brine to impinge on the coils and increases their efficiency.

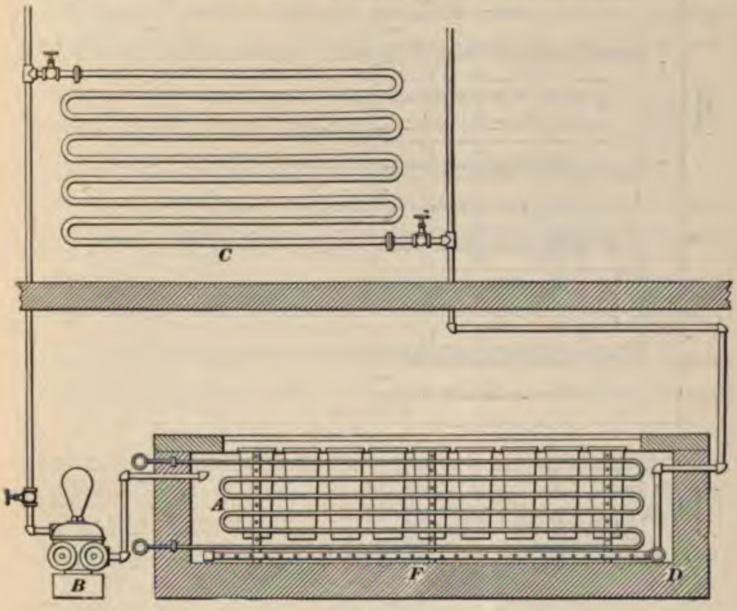

FIG. 365.

A propeller arrangement is shown in Fig. 366. The propeller *B* is driven by a belt or direct-connected engine at *A* and runs in a casing *E*. Wood partitions are built in the tank between the rows of cans and along the expansion coils *C*, *C*, *C*, as shown at *F*, *F*, *F*, the casing being made fast in the end of the partition. When the propellers are in operation, the brine is circulated as shown by the arrows. A 12-inch

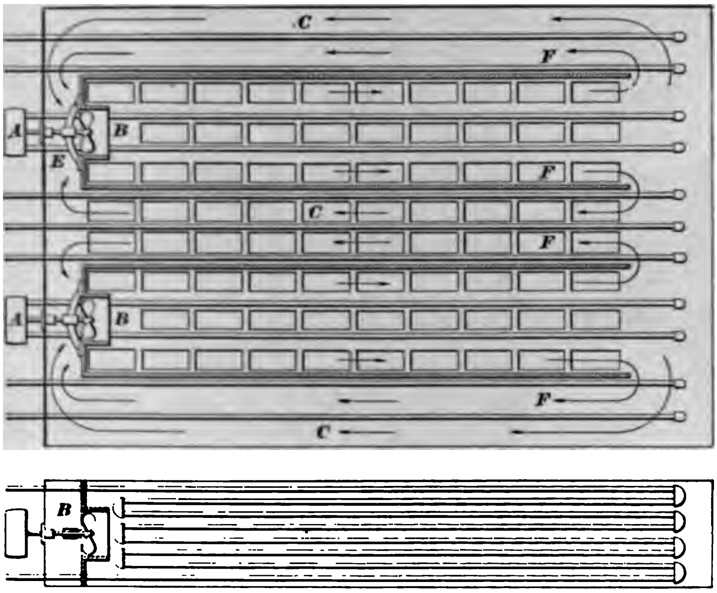

FIG 366.

propeller will suffice for a 10-ton ice tank and an 18-inch propeller for a 15-ton tank. This arrangement makes a very efficient method of circulating the brine, though it requires more power to operate it than either of the two systems described above.

1575. Crane and Hoist.—Ice plants are usually provided with hand cranes of the traveling type. A crane of this kind consists of a light channel-iron frame on which moves a 4-wheel trolley provided with a geared hoist. On the drum of the hoist, a rope or chain is run, and to this is fastened a can latch, a common form of which is shown in Fig. 367. The latches *A*, *A* are provided with hooks on the under side; these hooks engage into the holes in the sides of the can *B*. When the ends of the latches are moved in the direction of the arrows, the hooks disengage and the can is released. With a traveling crane and a geared hoist, one man can handle from 15 to 20 tons of ice in 12 hours. If, however, the plant is larger in capacity and it is desired to economize labor, a pneumatic hoist is recommended in place of the geared hoist. Pneumatic hoists are now made with a latch attached to the hook of the hoist that raises two cans from the tank at once. With such a device, one man can pull from 40 to 50 tons a day, or more than double the quantity he can pull with a geared hoist.

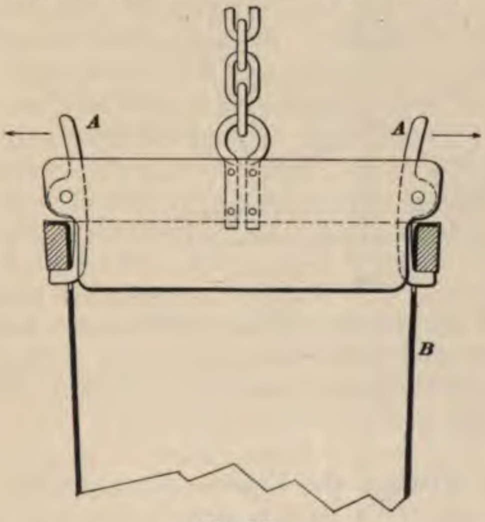

FIG. 367.

1576. Can Dump and Filler.—The can dump, or tilting frame, is shown in Fig. 368. It consists of a drip pan or tank *A*, to which are riveted fast a pair of supporting brackets *B*. These supporting brackets are provided with a hollow trunnion *C*, to which the water connection *D* is made. The box *E* is hung on the trunnions, and in it is placed the ice can *F*. The spray pipes *G* are also connected with the trunnion *C*. The arrangement of the trunnion is such that when the tilting box *E* is in the position shown by the full lines, no water enters the perforated pipes *G*; but when it is rocked on the trunnions and takes the position *E'*, the water is automatically admitted into the pipes *G*, from which it flows over the ice can and thaws the ice out of it. The box *E* is so weighted that when the ice leaves the can, the box returns by itself to the first position and shuts the water off automatically. In some of the smaller plants, a dipping tank is used, which is kept partially filled with warm water. The can is immersed in this tank and then drawn out again, the ice being freed by this means.

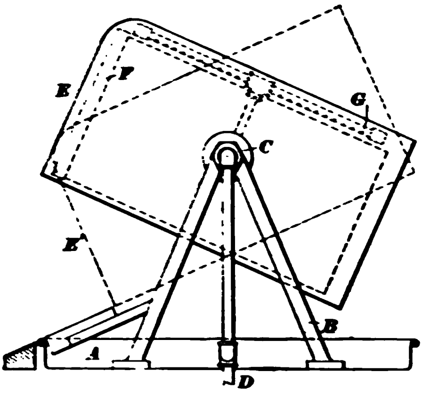

FIG. 368.

1577. Filling the Cans.—After the ice is dumped out of the can, the latter is brought back to its original place in the tank. It is then necessary to refill it with water for freezing. This is accomplished by means of an automatic can filler. A hose is connected with the filters of the distilled-water system, and to this hose the can filler is attached. After the can is placed in its position in the ice tank, the can filler is inserted and a trigger with which it is provided is pulled. This starts the water running into the can. When the water has reached the desired level in the can, a ball float strikes the trigger and shuts off the water-supply. In this manner, any number of ice cans may be filled, the operation requiring little attention. All the cans are filled to exactly the same level.

1578. Ice-Storage Piping.—In an ice plant, it is advisable to have an ice-storage room that will hold a two or four week run, as this will permit the plant to be started several weeks in advance of the ice season, so that there may be a good supply on hand when the season opens. Then it will be possible to shut down the machine in case it is necessary to make slight repairs, and the increased demand during very hot weather may be accommodated. Care should be taken to properly insulate the room, to prevent radiation as much as possible. It should be sufficiently high to store two tiers of ice on end, and the piping that keeps the room cool should be arranged overhead, with a space of at least 2 feet between the top of the upper tier and the brine pipes. The walls should be furred out to a distance of 6 or 8 inches, and slats should be nailed on the furring pieces to prevent the ice from coming in contact with the warm side walls. In case 1¼-inch pipe is used for the piping, 1 lineal foot of pipe will suffice for about 10 cubic feet of space. These rooms should be kept at a temperature of about 28°, no lower temperature being advisable, as the ice is liable to check or honeycomb. The ice should be placed on end and a space of about ½ inch left between the cakes, and slate or boards should be placed over the first layer before the second is put in. The temperature of the ice-storage room should not be allowed to go above 30°, as the ice is liable to melt, and if this happens, the cakes will freeze together when the room cools down again and it will be necessary to quarry out the ice.

1579. Water-Supply.—An ice plant should be provided with an independent water-supply, such as a well, river, or lake, and have its own water-pump *X* (Fig. 363) for the purpose of circulating the condensing water. The water is usually piped first to a receiving tank *T*, from there to the ammonia condenser *G*, then to the steam condenser *J*, to the boiler-feed pump *B*, from which it is passed through the feed-water heater to the boiler *A*. A separate supply of water direct from the pump is usually taken to the distilled-water cooler *U*. With the water at about 60°, 4 or 5 gallons per minute are usually allowed for each ton

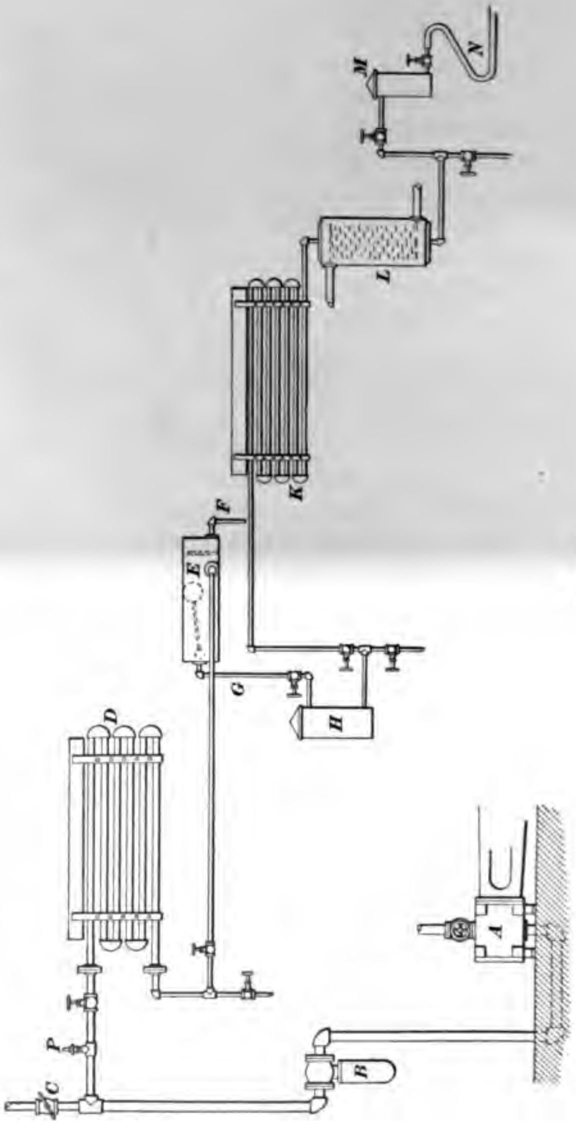

FIG. 369.

of ice made in 24 hours. In case the water is warm, say 80° or over, 6 to 8 gallons per minute should be allowed.

1580. Distilled-Water Apparatus.—The distilled-water system consists of the following parts:

(*a*) Oil separator.
(*b*) Back-pressure, or relief, valve.
(*c*) Exhaust-steam condenser.
(*d*) Reboiler and skimmer.
(*e*) Hot filter.
(*f*) Cooling coil.
(*g*) Gas for cooler.
(*h*) Cold filter.

The usual process of distillation is as follows (see Fig. 369): The exhaust steam on leaving the engine passes to the oil separator *B*. Here all the oil and any priming water that the steam may contain are separated out. The steam then passes to the exhaust-steam condenser *D*. Any surplus steam that is not immediately condensed in *D* is taken care of by the back-pressure valve *C*, which is usually set at about 2 or 3 pounds above the atmosphere. A small vent-cock is also provided in the pipe at *P* for the purpose of relieving the steam-pipe of any air or other extraneous gases. The condensed steam then passes to the reboiler *E*, where it is reboiled by means of live steam. The reboiler is arranged with a skimming diaphragm, so that any light impurities may float to the surface and be skimmed off and go to waste through the pipe *F*. The reboiled condensed steam now passes through the pipe *G* to the hot filter *H*, whence it goes to the distilled-water cooler *K*. Here it is cooled down to the temperature of the condensing water and then enters the top of the forecooler *L*. In this vessel it is cooled down to a temperature of 45° or 50°; then it passes through the cold filter *M*, and finally to the cans through the hose *N*. The object of distillation is, in the first place, to eliminate all oil and other foreign matter, and then to expel any air or gases that may be contained in the water when it enters the boiler. Most of these gases are blown out of the cock *P*, those remaining being driven off in the reboiler *E*.

1581. Oil Separator.—It is very important to get a first-class oil separator on the exhaust-pipe. The separator should be made the full size of the exhaust-pipe, as a smaller separator is insufficient to separate out all the oil carried by the steam.

1582. Exhaust-Steam Condenser.—The exhaust-steam condenser is usually of the atmospheric type; it is made of 2-inch pipe, enough coils being used to give an area of 1½ to 2 times the area of the main exhaust-pipe. Enough water is run over these coils to condense the steam, but not sufficient to cool the condensed steam to any extent. About 8 square feet of surface per ton of ice made per 24 hours is a good proportion.

1583. Reboiler and Skimmer.—The reboiler and skimmer is shown in detail in Fig. 370. This form is commonly used and is quite popular, as it works well and is cheap. It consists of a rectangular tank *A*, which is provided with a heavy galvanized-iron diaphragm *B*, reaching to within 3 or 4 inches of the top of the tank. The end of the tank nearest the diaphragm is provided with a pipe flange *D* for taking off the water that is skimmed over the diaphragm. The water inlet at *C* is connected with the exhaust-steam condenser and is provided with a perforated pipe *E* inside

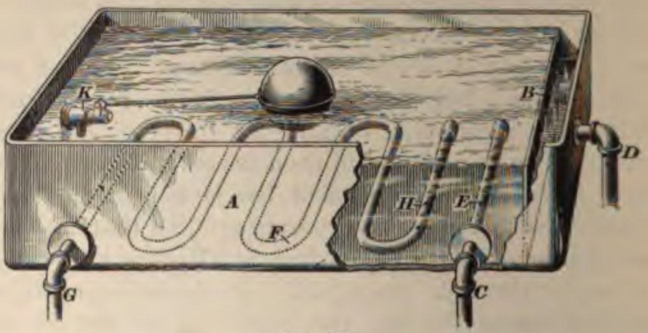

FIG. 370.

the tank. The water coming from the condenser enters through this pipe and is distributed across the whole width of the tank. Any trace of oil that may have escaped the separator will float to the top and be skimmed off by means of the diaphragm *B*.

The reboiling is accomplished by means of the galvanized coil *F*, which is connected with the live-steam pipe at *G* and is provided with perforated openings at *H*. The live steam entering *G* heats the water near that end of the tank to the boiling point and keeps it boiling throughout the whole length of the coil; the condensed live steam passes out through the perforations in *H*. This allows the whole body of distilled water that is contained in the tank *A* to be thoroughly freed from air, etc., while it is passing from the inlet *C* to the outlet *K*. The outlet *K*, which connects with a hot filter, is provided with a float-valve, which is so arranged that when the tank is full the valve is wide open; it shuts gradually as the water level drops in the tank *A*, and entirely closes when the level is some 2 or 3 inches above the valve. This arrangement prevents the distilled water from being charged with air, as the water level can not drop down to that of the outlet, and permits the skimmer to work during the time that the can filler is not in use and when the tank has reached its level and is overflowing. All parts of the distilled-water apparatus, including coils, etc., should be heavily galvanized or else made of copper or brass.

1584. Filters.—The filters are usually heavy sheet-iron or cast-iron galvanized cylinders provided with a perforated false bottom. On this perforated bottom the filtering material is placed. In ice making, filters usually filter from top down, though some engineers prefer to filter from bottom up. All connections should be made in the side of the cylinder, so that the covers can be readily removed for cleaning or recharging. Stop-cocks or valves should also be provided on the distilled-water connections from and to the filters, and by-pass connections should be made so that the filter can be shut off, the by-pass opened, and the running of the plant not interfered with when the filter is being recharged or cleaned. As the distilled water that is used in an ice plant is comparatively free from impurities, the filters do not have much work to do, and it is usually sufficient to charge a filter once a season. For this reason, wash-out filters of different patent makes are seldom used in distilled-water plants.

When the filter is recharged, it should be thoroughly rinsed with distilled water so as to drive out all air and wash the filtering material.

It is advisable to let the distilled water run to waste for some time before filling any cans, so as to be sure that all air and other gases are eliminated.

For hot filters, crushed quartz is found very efficient. For cold filters, quartz, sand, or maple charcoal may be used. Filters between the forecooler and the ice tank are found very effective for eliminating certain red cores in ice, as this red does not precipitate except at a temperature very near the freezing point.

For either sand or quartz filters, 1 square foot of filtering surface is sufficient for a 15-ton ice plant. In filters that filter from top down, if the cans are found to fill slowly, the cover of the filter can be removed and the top of the filtering material skimmed off with a shovel, which will remove with it most of the foreign matter without the necessity of recharging the filter.

1585. Water-Cooling Coils.—The cooling coils are built like the steam condenser, but usually of pipes of a smaller size. The inlet is at the top of the coil and the outlet at the bottom. Four square feet of pipe surface should be allowed for each ton of ice.

1586. Gas Forecooler.—The gas forecooler consists of a cylindrical or rectangular tank, in which are placed galvanized-iron pipe coils. These coils are connected at the top and bottom with the compressor suction-pipe and the expansion coils from the tank, respectively. The cold gas leaving the expansion coils in the ice tank passes through these coils and cools the water contained in the body of the forecooling tank. The area of these coils should in no case be less than that of the main suction-pipe of the machine. The best practice makes them from 1½ to 2 times the area of that pipe. The coils should be kept well away from the body of the tank, so that ice may form on them without freezing against the walls of the tank and bursting it. The tank should be provided with a cover and should be air-tight, as it is subjected to the pressure of the water from the reboiler.

The usual allowance for forecooling coils is 4 to 5 square feet per ton of ice.

A forecooler added to the plant allows the water to come to the ice cans at a temperature of 40° to 50° and increases the capacity of the plant considerably.

1587. Distilled-Water Connections.—All connections and shells of all vessels should be either of block tin or galvanized iron. All valves should be of composition, no iron-body valves being used, after the exhaust steam once enters the steam condenser. The various parts of the distilled-water system should be connected with live-steam pipes, so as to permit of their being blown out and thoroughly cleansed. Blow-off cocks should be provided at short intervals on the various pipes, and all filters and other closed vessels should be provided with them. If the distilled water begins to run slowly into the cans, this indicates that the filters need cleaning or recharging.

ABSORPTION ICE-MAKING PLANT WITH CAN SYSTEM.

1588. General Description.—The absorption machine is well adapted for making ice, as it does not produce oil in the exhaust steam as does the compression machine. The usual arrangement is to take the condensed steam that leaves the generator to the reboiler, then pass it through the water-cooler to the filters, and finally to the ice cans. This arrangement makes it unnecessary to use any oil traps, skimmers, etc.

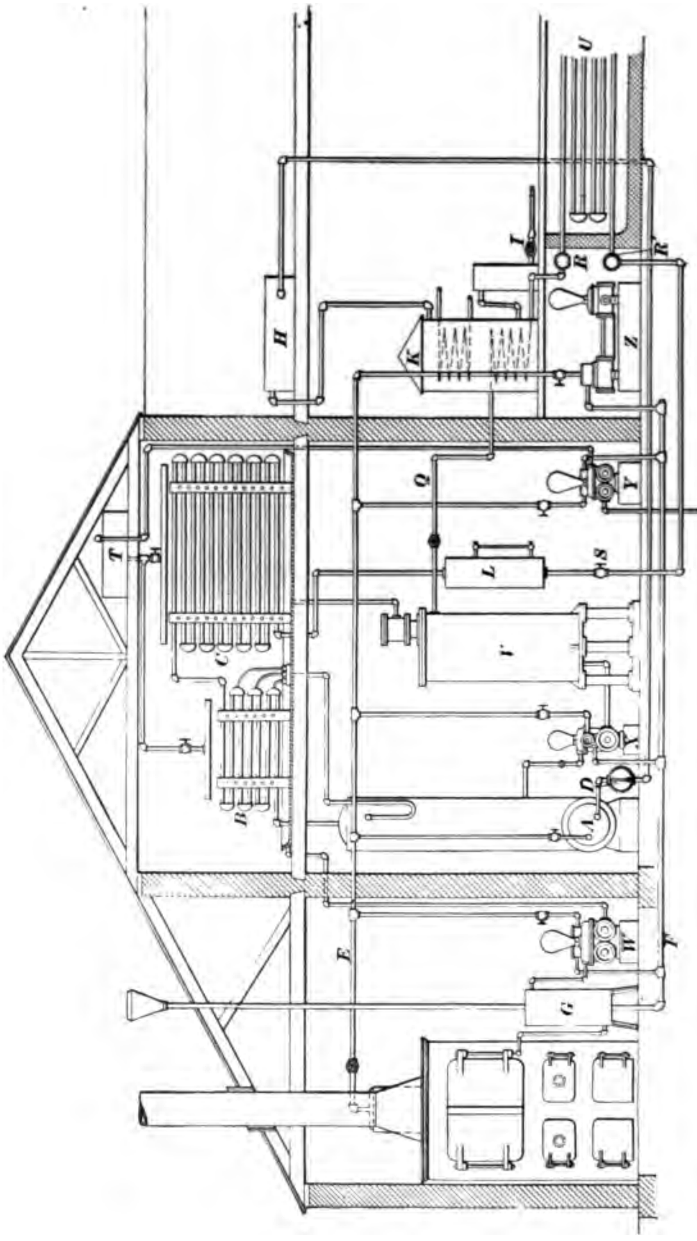

FIG. 371.

The general arrangement of an absorption ice-making plant is shown in Fig. 371. The steam main *E* leading from the boiler furnishes steam to the generator *A* and pumps *W*, *X*, *Y*, and *Z*. The exhaust from the four pumps is led to the main *F*, and thence passes through the feed-water heater *G*. The steam from the generator passes to the small receiver *D* and from there to the reboiler *H*. The course of the ammonia may readily be followed; the gas from the generator passes to the rectifier *B*, thence to the condenser *C*. The liquid gathers in the receiver *L* and is admitted to the brine coil *U* through the expansion-valve *S*. The gas from the coil *U* is carried by the pipe *Q*, which passes through the cooler *K*, to the absorber *V*. The pump *X* takes the liquor from the absorber and delivers it to the exchanger, which in Fig. 371 is hidden behind the analyzer. The student will find it advantageous, in tracing the course of the gas, to compare Fig. 371 with Fig. 351.

The pump *Y* takes water from some source, as a well or city water-supply, and delivers it to the tank *T* in the top of the building. From this tank it flows over the rectifier and condenser into a pan below them. From this pan is drawn the water for the boiler-feed pump *W* and for the absorber *V*. The pump *Z* is used to circulate the brine.

1589. The Distilled-Water System.—The course of the distilled water is more clearly shown in Fig. 372. The steam enters the generator *A* through the pipe *B*, and after condensing, passes out at *C* to a small receiver *D* having a gauge-glass *E* at one end. The outlet of this receiver is at the bottom; from it a pipe *F*, which can be throttled at *G*, leads to the reboiler *H*. Here the distilled water is thoroughly reboiled, then it passes successively to the cooler *K*, the filter *I*, and finally to the ice cans.

The upper portion of the distilled-water cooler *K* is fitted with a helical coil *N*, whose tails *L* and *M* project through the side of the tank. The gas-forecooling coil *O* is in the lower portion of the tank, with its tails *P* and *Q* passing through the shell, as shown. The water used for cooling the distilled water enters the bottom of the coil *N* through the tail *M* and passes out at *L*. The absorber suction-pipe *Q* is connected with the upper tail, and the return-gas header *R* of the expansion coil is connected with the lower tail of the forecooling coil *O*.

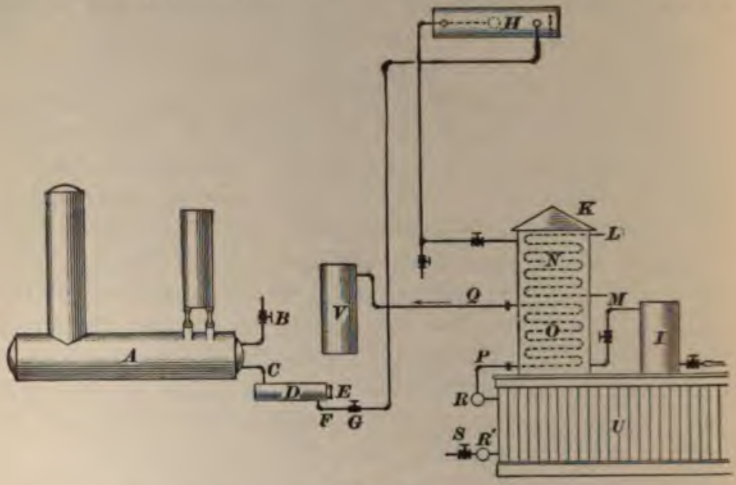

FIG. 372.

The method of operation is as follows: After the machine has been started and distilled water begins to accumulate in the receiver *D*, as shown by the glass *E*, the valve *G* is opened slightly so as to keep the level of the liquid in *E* constant. This permits the distilled water to enter *H* and pass to waste over the skimming diaphragm. After the mass of water in *H* has become thoroughly heated up and reboiled, it is allowed to follow the course above described to the ice cans.

1590. Operating the Plant.—When a can plant is first started, live steam should be allowed to blow for several hours through the various parts of the distilled-water system. The distilled water is then turned on and allowed to run to waste for several days before any cans are filled. As the plant is designed to keep the brine at a certain temperature while freezing the water in the cans, and as it is the heat given off by the water in freezing that prevents the brine from falling below that temperature, it is advisable not to run the plant with the cans empty, as otherwise the temperature may drop so much as to freeze the brine. In order to have the cans full when the plant is first started, ordinary city or well water may be used to fill them with, as it is of course impossible to get distilled water before the machinery has been running for some time. The ice from this water is very poor and need not be used except in case of necessity. It takes from 4 to 5 days to freeze the batch of ice from this water, but the pulling of the ice may be begun at the end of the third day even though the cakes are not thoroughly frozen. Then the cans are filled with distilled water.

1591. The capacity of an ice plant is governed more particularly by the number of cans filled than by those pulled in a given time. Some unscrupulous builders of ice machinery, taking advantage of this fact, freeze up the initial, or first, batch of ice, which gives them as a start 2½ times the nominal capacity of the plant. The plant is then operated for a week and is able to turn out from 5 to 10 per cent. more ice than its nominal capacity. This is due to the fact that they start with a tank full of ice and end up with one nearly empty. A 30-day run is the minimum time on which the capacity of an ice plant should be based.

1592. If it is found necessary to shut a can plant down for any length of time, care should be taken to see that all cans in which the ice has freed itself from the side of the can and floated up are emptied and refilled with fresh water. If this is not done, the cans will be bulged and the bottoms may burst when the plant is started again. In the case of an absorption plant, the generator coil should be thoroughly blown out every 24 or 36 hours so as to eliminate all rust, etc.

PLATE ICE PLANTS.

1593. The Two Systems Used.—There are two systems of plate ice plants, viz.: the **dry-plate** and the

wet-plate system. In the dry-plate system, the ammonia gas is expanded into a return-bend coil, such as is shown in Fig. 373, the space between the runs being filled in with wood or iron, so as to separate the two cakes forming on the sides of the coil. In the wet-plate system, the freezing is accomplished by means of a tank or cell several inches thick, in which brine circulates. This tank either contains the direct-expansion coil or else the brine is piped from a separate cooling tank or brine cooler to the cells. In either case the ice adheres to the side of the plate, and when the right thickness is reached, it is thawed off by running either warm brine or water into the cells, or, in the dry system, by turning the hot ammonia gas direct from the condenser into the coils.

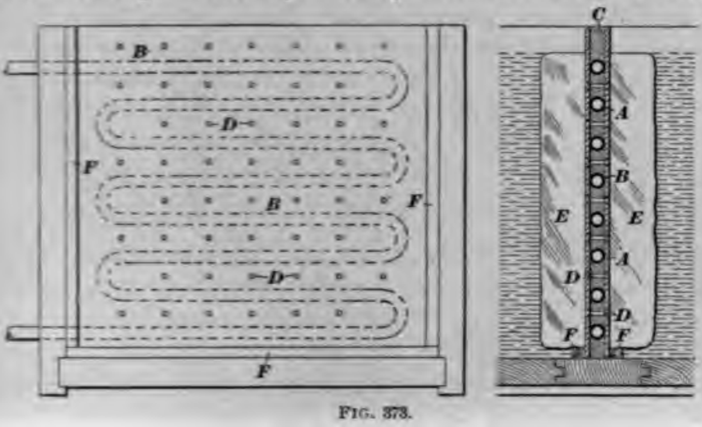

FIG. 373.

1594. Dry-Plate System.—The dry-plate system most commonly used is that shown in Fig. 373. It consists of a continuously welded return-bend or zigzag coil *B*, between the runs of which are placed blocks of wood *C*. On each side of this coil are laid plates of sheet steel *A*, *A*, and these are bolted or riveted through by means of the bolts or rivets *D*. This makes a comparatively inexpensive arrangement, and a plant constructed with such plates is not much more. expensive than a can plant. These plates, which are about 30 inches on centers, are placed in a long wooden tank. Water is run in between the plates to within about 9 inches from the top, and the expansion ammonia is then turned on. Great care must be taken, in feeding this kind of plate, that the freezing does not take place too rapidly, or the air-bubbles will not have time to disengage and white ice will be the result. After a certain amount of ice has accumulated on the coils, say about 4 inches, care must be taken not to feed too rapidly, or there is a liability of the coil frosting through and liquid ammonia getting over to the compressor.

The great objection to a dry plate is that if the expansion is insufficient, the ice will not form evenly over the whole plate, but will be thicker where the ammonia is fed in, gradually tapering down to a thin cake at the outlet. Plate plants of this type usually have forecooling coils of large surface, so as to take up any excessive expansion.

On the whole, a dry-plate plant is very difficult to operate with any degree of success.

1595. Wet-Plate System.—A wet-plate system very commonly used is shown in section in Fig. 374. It consists of wrought-iron cells filled with brine, in which are placed the direct-expansion coils. The wrought-iron cell *B B* contains the ammonia expansion coil *A* and is filled with brine. The ice *E E* forms on the sides of the cells. With this arrangement, it is possible to get a very clear ice out of ordinary well, river, or city water without the necessity of distillation. The ice forms gradually on the sides of the plate, and the temperature of the brine can always be kept at any desired point by the proper

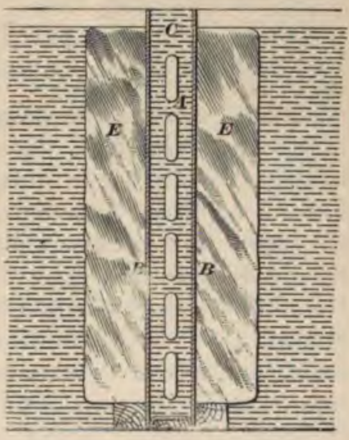

FIG. 374.

manipulation of the expansion-valve. After a little practice, it is easy to ascertain just the temperature of brine that is necessary for ice of different thicknesses. This has to be determined by experiment, as no two samples of water contain the same ingredients.

The principal objection to the wet-plate system is its large first cost and the liability of the cells leaking. If the cells are not properly made in the first place, the expense of maintaining them is considerable, the smallest leak of brine injuring the quality and appearance of the ice.

1596. Plate Plant Apparatus.—A plate ice plant usually consists of the following parts:

(*a*) Ice tank with plates.
(*b*) Forecooling tank with coils.
(*c*) Crane with hoists.
(*d*) Tilting table.
(*e*) Cutting-up saws.
(*f*) Water filters.
(*g*) Water connections.

In case of a wet-plate system, there are also

(*h*) Brine-storage tank.
(*i*) Brine-transfer pump.
(*j*) Brine connecting mains.

1597. Lifting and Cutting the Plates.—In order to lift the plates from the tank, two tension bolts, which have eyes at the top to which the crane is fastened, may be frozen in the ice. In other plants, a chain is slipped under the cake. This is accomplished by first passing a copper wire under the cake, after the latter, having been detached from the plate, floats to the surface; then the chain is made fast to one end and then drawn up around the cake by means of the wire. After the cake is drawn from the cell, it is necessary to lay it down. This is done by means of the tilting table *A*, Fig. 375, which is tilted by means of a worm-gear. The table is placed in a position such as shown by the full lines. The crane *D* is then run over and the cake *C* is allowed to lie on this table. The chains are taken off, and the table is then lowered into the position shown by the dotted lines.

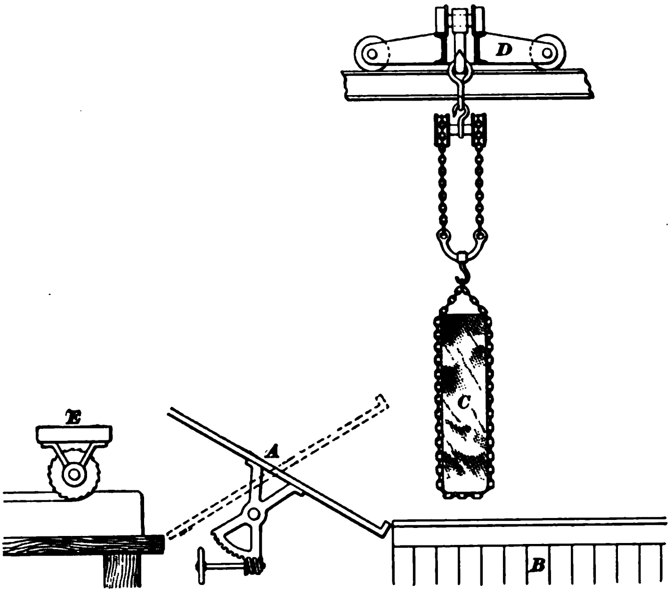

FIG. 375.

It is now necessary to cut the cake into merchantable-sized ice, which is usually accomplished in large plants by a gang of steam saws *E*. These are driven by power and the cake is fed through under them. The saws cut a groove about 3 inches deep in the cake, which is then split by hand with chisel bars. In small ice plants, what is known as a steam ice-cutter is used. This consists of a blade of steel with an 8-inch copper pipe brazed fast to one end. This is placed on edge on top of the cake of ice *F* to be cut, as shown in Fig. 376, the steel blade *A* being held vertical and steam turned on into the copper pipe *B*. This rapidly thaws the ice and cuts down through. In other plants, an ordinary hand-saw is used to score the ice and chisel bars to separate the cakes.

The method of thawing the ice in a wet-plate plant is as follows: When the ice in a cell has reached very nearly the required thickness, the feed valve to that cell is shut off, but the suction-valve is left open. This draws all the ammonia gas from the coil. The thickness of the ice is also increased thereby ½ to ¾ inch. The ice is now ready for thawing. The brine contained in the cell is drawn out by means of the brine-pump and the cell is filled with cold water. Water warmer than 60° should not be used, as the ice is liable to shiver. If, however, the thawing water gets as low as 36°, it is best to run it off into the draining tank and refill the cell. After the water has stood in the cell for 4 or 5 hours, a chisel bar can be inserted along the top between the cake and the cell. If it is thawed free of the cell, the cake when gently pried will float up in the water. The crane is then made fast and the cake is drawn out. The thawing water may now be run out of the cell, and the ice space may be thoroughly cleansed of any dirt that may have been precipitated during the process of freezing by opening the waste-cock and washing the cell clean with a hose. The cell is then filled with brine and the tank with water to within 9 inches of the top; the expansion is turned into the coil, and the process of freezing is started again.

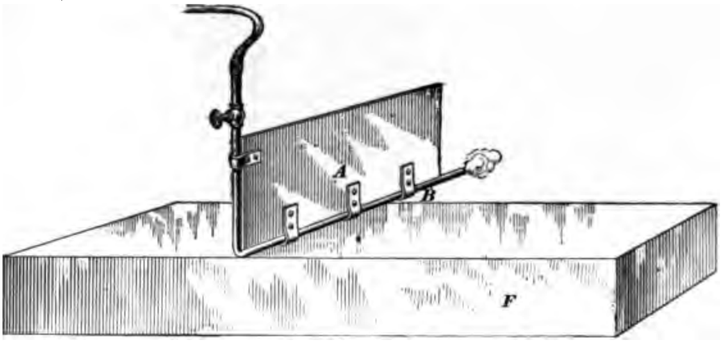

FIG. 376.

MISCELLANEOUS USES OF REFRIGERATION.

1598. Hotel Refrigeration.—The work required in hotels is the refrigeration of a large number of small boxes used for various purposes about the establishment, such as wine and beer boxes, general storage, meat, cook stores, oysters, fish, ice cream, etc.; also the freezing of caraffes and the making of small quantities of ice. Such work is out of all proportion to the space cooled, owing to the fact that the boxes are small and usually scattered throughout the building. For small refrigerating boxes for these purposes, namely, those of 100 cubic feet or less, about 500 cubic feet should be allowed for each ton of refrigerating effect. For boxes ranging between 100 and 1,000 cubic feet, 1 ton will suffice for about 1,200 cubic feet of space. In addition to this, the amount of ice that is to be made should be carefully considered, 2 tons of refrigerating effect being allowed for every ton of ice made.

Direct expansion is out of the question for hotel refrigeration, on account of the large number of boxes or rooms that would demand attention.

In calculating the capacity required for hotel refrigeration, it is always better to err on the safe side and get a machine rather too large than too small.

1599. Chocolate-Factory Refrigeration. — The setting of chocolate requires a cool, dry atmosphere in the cooling rooms. These rooms are usually kept at a temperature of about 68°. The indirect system of cooling and ventilating should be used, so as to prevent the possibility of any moisture from drip coils, etc. After allowing for the amount of refrigeration required to cool the given space to 68°, the number of persons in the coating room and the amount of chocolate to be cooled daily should be considered. One ton of refrigeration will be required to absorb the heat given off by 7 persons. It takes about 1,200 B. T. U. per pound to condense and freeze the moisture in the atmosphere, of which a great deal is exhaled by the persons in the room. The conveying of the cold air from the bunker and the taking away of the vitiated air from the rooms should be effected by means of a series of air-ducts or flues, the exact size and location of which should be determined by some expert ventilating engineer. This is quite important, in order to prevent drafts and the consequent colds to the persons in the coating room.

1600. Air Refrigeration.—Air is often cooled for various purposes, such as drying photographic plates, etc. When the object of the cooling is to dry the air, the amount of refrigeration required depends largely upon the amount of moisture that has to be eliminated by condensing and freezing it, the cooling of the air itself being of secondary importance. As stated in the preceding paragraph, 1,200 B. T. U. may be allowed for every pound of moisture taken from the air, in addition to the refrigeration required to cool the air to the temperature of condensation of the moisture it contains.

A SERIES

OF

QUESTIONS AND EXAMPLES

RELATING TO THE SUBJECTS
TREATED OF IN VOL. II.

It will be noticed that, although the various questions are numbered in sequence from **504** to **794**, inclusive, these questions are divided into five different sections, corresponding to the five sections of the preceding pages of this volume. Under the heading of each section are given, in parenthesis, the numbers of those articles which should be carefully studied before attempting to answer any question or to solve any example occurring in the section.

PNEUMATICS.

(ARTS. 1039–1088.)

(504) What do you understand by *tension of gases?*

(505) A cylinder filled with compressed air supports a column of mercury 4 feet high. (*a*) What is the tension of the air in pounds per square inch? (*b*) In atmospheres? Take the weight of a cubic inch of mercury in all cases as .49 pound.

Ans. (*a*) 23.52 lb.
(*b*) 1.6 atmos.

(506) By reason of a partial vacuum, a column of water 19 feet in height is supported by the atmosphere. (*a*) How many inches of vacuum does the gauge show, and (*b*) what is the pressure above the mercury in pounds per square inch?

Ans. (*a*) 16.828 in.
(*b*) 6.454 lb. per sq. in.

(507) A closed vessel, fitted with a piston, contains coal gas under a pressure of three atmospheres. If the piston is so moved that the volume is 2½ times its former volume, what is the tension of the gas in pounds per square inch? The temperature is the same in both cases.

Ans. 17.64 lb. per sq. in.

(508) A certain quantity of air, under a pressure of 1½ atmospheres and a temperature of 75°, weighs 7.14 pounds. It is put in an empty vessel in which one cubic foot of air weighs .08 pound. (*a*) What is the new volume? (*b*) The temperature, the pressure remaining the same? (*c*) The original volume?

Ans. (*a*) 89.25 cu. ft.
(*b*) 283.887°.
(*c*) 64.188 cu. ft.

(509) The temperature of the discharged air of an air compressor, the tension of which is 40 pounds per square

inch, is 120°; when it has cooled down to the temperature of the surrounding air, which is 55°, what is its tension?
Ans. 35.517 lb. per sq. in.

(510) What is the weight of a cubic foot of air at 60°, under a pressure of one atmosphere? Ans. .076296 lb.

(511) The total pressure upon a body is 175,000 pounds per square foot; what is the equivalent pressure in atmospheres? Ans. 82.672 atmos.

(512) Three gases, oxygen, hydrogen, and nitrogen, are mixed together in a vessel containing 40 cubic feet. The volume and tension of the oxygen are 12 cubic feet and one atmosphere, respectively; of the hydrogen, 10 cubic feet and two atmospheres; of the nitrogen, 8 cubic feet and three atmospheres. The temperature of the separate gases and of the mixture remaining the same throughout, what is the tension of the mixture? Ans. 20.58 lb. per sq. in.

(513) In the last example, suppose the volume of the mixture is not known, and that the tension is required to be 23 pounds per square inch; what is the volume of the mixture? Ans. 35.79 cu. ft.

(514) A balloon is filled with 10,000 cubic feet of hot air at a temperature of 280°. If the temperature of the surrounding air is 77°, and the weight of the balloon and fixtures is 100 pounds, how great a weight will it lift? The tension of the hot air is one atmosphere. Ans. 102.68 lb.

(515) A vessel containing 13 cubic feet of air having a temperature of 73° and a tension of one atmosphere is placed in communication with another vessel containing 18 cubic feet of air, at a temperature of 53° and a tension of 30 pounds per square inch. What is the new temperature if the tension of the mixture is 20 pounds per square inch?
Ans. —20.65°.

(516) What is a vacuum? Illustrate it.

(517) An air pump produces a vacuum of $\frac{1}{40}$ of an inch; what is the pressure upon a square foot? Ans. 1.764 lb.

(518) A horizontal cylinder, closed at one end and open

at the other, is exactly fitted with a piston having a hole in it to allow the confined air to escape. The length of the cylinder is 6 feet, and the diameter is 40 inches. The piston weighs 325 pounds, and is moved inwards until the length of the space between the cylinder head and the piston is 40 inches. (*a*) How great a force will be necessary to pull the piston out of the cylinder after the hole has been plugged, if the coefficient of friction is 14%? (*b*) To shove it in until the length of the enclosed space is 6 inches? Assume that the temperature remains constant.

Ans. { (*a*) 8,255.55 lb.
(*b*) 104,723.612 lb.

(519) There are 8.47 cubic feet of air, under a pressure of 38 pounds per square inch. If 4½ cubic feet be removed, what will be the tension of the remainder, the temperature remaining the same? Ans. 17.812 lb. per sq. in.

(520) A vessel containing 3 cubic feet of gas weighing .5 pound under a pressure of one atmosphere has compressed into it enough more of the gas to make it weigh 1 pound and 6 ounces; the temperature remaining the same, what is the new tension of the gas in pounds per square inch? Ans. 40.425 lb. per sq. in.

(521) If 4,516 cubic inches of gas having a temperature of 260° are cooled down to a temperature of 80°, the pressure remaining the same, what is the volume?

Ans. 1.96 cu. ft.

(522) If 55 cubic feet of air, under a pressure of 1¼ atmospheres, have a temperature of 88°, what is the weight?

Ans. 4.977 lb.

(523) Two vessels, the volumes of which are each 7½ cubic feet, are filled with air; the temperature is the same in both, but the tension in one is two atmospheres, and in the other 40 pounds per square inch. If all of the air in one vessel is compressed into the other, what is the tension of the mixture after it has cooled down to the original temperature? Ans. 69.4 lb. per sq. in.

(524) A solid block of wood, 48″ × 36″ × 24″, weighs in air 1,200 pounds; how much will it weigh in vacuum? The temperature of the air is 60°. Ans. 1,201.83 lb.

(525) A double-acting steam pump is required to force water to a height of 127 feet; the height of the suction is 16 feet. Allowing 25% for friction, etc., (*a*) what must be the horsepower of a steam engine to drive this pump, if the diameter of the plunger is 9 inches, stroke 12 inches, and number of strokes per minute 125? (*b*) How many gallons could be discharged per hour?

Ans. $\begin{cases} (a)\ 19.942 \text{ H. P.} \\ (b)\ 24{,}784.3 \text{ gal.} \end{cases}$

(526) In the last example, if the pump is single-acting, and the number of strokes per minute 100, what will be the discharge in gallons per hour? Ans. 9,914.4 gal.

(527) If you are told that the vacuum gauge of a condenser shows 23 inches vacuum, what do you understand by it? What is the pressure in the condenser?

(528) What is a pressure of one atmosphere equivalent to in pounds per square foot? Ans. 2,116.8 lb. per sq. ft.

(529) If the weight of 3 cubic feet of air at a certain temperature and under a pressure of 30 pounds per square inch is .27 pound, what is the weight of 1 cubic foot under a pressure of 65 pounds per square inch and at the same temperature? Ans. 0.195 lb.

(530) In the last example, what is the temperature of the air? Ans. 439.6°.

(531) What are the absolute temperatures corresponding to 32°, 212°, 62°, 0° and − 40°?

(532) Three and one-half pounds of air, under a pressure of 10 atmospheres, occupy a volume of 4 cubic feet. What is the temperature? Ans. − 6.583°.

(533) Fifteen cubic feet of oxygen, having a tension of 63 pounds per square inch, and 19 cubic feet of nitrogen, having a tension of three atmospheres, are mixed together in a vessel the volume of which is 25 cubic feet. The temperature of both gases and of the mixture being the same, what is the tension of the mixture? Ans. 71.316 lb.

(534) One pound of air has a temperature of 80° and a volume of 10 cubic feet; what is its tension?
Ans. 20 lb. per sq. in.

(535) If an indicator card taken from a condensing engine shows a pressure of 12½ pounds below the atmosphere, how many inches of vacuum will the vacuum gauge show?

(536) A vacuum of 27 inches will support a column of water of what height? Ans. 30.6 ft.

(537) A certain vessel has a volume of 6.7 cubic feet. A vacuum gauge attached to it shows 17½ inches. (*a*) How much air at atmospheric pressure will it be necessary to admit to have the vacuum gauge show 5 inches? (*b*) to show 0 inches? Ans. (*a*) 2.79⅙ cu. ft.

(538) A certain vessel contains 11 cubic feet of gas weighing 2.4 pounds. If put in communication with a second vessel from which all the air has been removed, and which has a volume of 25 cubic feet, what will be the weight of a cubic foot of the gas, the temperature remaining constant?
Ans. $\frac{1}{15}$ lb.

(539) The air contained in a closed vessel, under a pressure of 12 pounds per square inch, is heated from 60° to 300°; what is its tension? Ans. 17.54 lb.

(540) What is the weight of a cubic foot of air at 212°, under a pressure of one atmosphere? Ans. .059039 lb.

(541) The diameter and stroke of the piston of an air compressor is 20 inches and 32 inches, respectively. If the discharge valve opens when the piston has completed 26 inches of its stroke, (*a*) what is the volume? (*b*) the weight? (*c*) the tension of the air discharged? Take the temperature of the outside air as 75°, and the temperature at discharge as 125°.

Ans. {
(*a*) 1,884.96 cu. in. = 1.0908 cu. ft.
(*b*) .43143 lb.
(*c*) 85.727 lb. per sq. in.

(542) Nineteen cubic feet of air, having a tension of 12 pounds per square inch, are mixed in a vessel which holds 30 cubic feet, with 21 cubic feet of air from another vessel. If

the tension of the mixture is 35 pounds per square inch, what was the tension of the second volume of gas?

Ans. 39.14 lb. per sq. in.

(543) A vessel containing 45 cubic feet of air at a temperature of 60° and a tension of 13 pounds per square inch is emptied into another vessel containing 60 cubic feet of air at a temperature of 80° and a tension of 17 pounds per square inch. What is the tension of the mixture if its temperature is 72°? **Ans. 26.723 lb. per sq. in.**

(544) What is a *partial vacuum?* If enough air is admitted to the vacuum chamber to cause the column of mercury to be 4½ inches shorter than the barometer column, how many inches of vacuum will the gauge show?

(545) By reason of a partial vacuum, a column of alcohol 16 feet high is supported. How many inches will the vacuum gauge show? **Ans. 11.337 in.**

(546) The stroke and diameter of the piston of a blowing engine (one form of an air compressor) are each 80 inches. The valves are so set that they will open for discharge when the tension of the compressed air becomes 9 pounds above the atmosphere. (*a*) At what point of the stroke will the valves open? (*b*) How many cubic feet of air having this tension will be discharged during one stroke of the piston, the temperature being constant throughout?

Ans. $\begin{cases} (a)\ 30.38 \text{ in.} \\ (b)\ 144.34 \text{ cu. ft.} \end{cases}$

(547) A certain quantity of air under a pressure of 3½ atmospheres weighs 13 pounds. After expanding under a constant temperature, the weight of the same volume is only 2 pounds. What is the tension of the air?

Ans. 7.915 lb. per sq. in.

(548) The stroke of the piston in an air compressor is 60 inches. When the piston has traveled 50 inches, what is the tension (the temperature at discharge being 130°) of the enclosed air, assuming that the delivery valves do not open until this point is reached? The original temperature is 60°. Ans. 100.07 lb. per sq. in.

(549) A pound of air has a temperature of 127° and a tension of 27 pounds per square inch. What is its volume? Ans. 8.055 cu. ft.

(550) The weight of a certain body of air having a tension of 4,000 pounds per square foot and a temperature of 100° is .5 pounds. What is its volume? Ans. 3.735 cu. ft.

(551) Forty cubic feet of air having a temperature of 100° and a tension of 90 pounds per square inch are mixed with 57 cubic feet having a temperature of 130° and a tension of 80 pounds per square inch. The tension of the mixture is 120 pounds per square inch and the temperature is 110°. What is the volume? Ans. 67.248 cu. ft.

(552) If a bottle fitted with a rubber bulb, as shown in Fig. 199, be filled with water until the air in the bulb occupies a space of 20 cubic inches under a tension of exactly one atmosphere, what will be the total pressure on the bottom of the bottle caused by squeezing the bulb until the space inside is only $\frac{1}{4}$ of its original volume? The opening in the neck of the bulb is $\frac{1}{4}$ of a square inch, the bottom of the bottle is 3 inches in diameter, and the depth of the water is 12 inches. Ans. 314.793 lb.

(553) Four cubic feet of air is heated under a constant pressure from 40° to 115°. What is the resulting volume? Ans. 4.6 cu. ft.

HEAT.

(ARTS. 1089–1188.)

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(554) If two pounds of a certain substance and one pound of water are heated under exactly the same conditions from 60° to 170°, and it takes 55 minutes to heat the substance to this point and 9 hours and 25 minutes to heat the water to the same point, what is the specific heat of the substance? Ans. .04867.

(555) A steel piston rod is 4 inches in diameter, and the piston is bored 3.9985 inches in diameter to receive it. To what temperature must the piston be heated, assuming its original temperature to be 80°, and the diameter of its bore after heating to be 4.001 inches, to allow the rod to enter freely? Ans. 184.4°.

(556) If the whole piston was raised to the same temperature as the bore, and its weight was 360 pounds, (*a*) how many units of heat were required, assuming a loss of 12% by radiation, etc.? (*b*) How many foot-pounds of work is this equivalent to?

Ans. { (*a*) 4,975.6 B. T. U.
(*b*) 3,871,017 ft. lb.

(557) The stroke of the piston of an air compressor is 80 inches, and its cylinder is 80 inches in diameter. If the air be compressed isothermally to a pressure of 120 pounds per square inch, (*a*) what will be the volume discharged, and (*b*) how much work will be required?

Ans. { (*a*) 28.507 cu. ft.
(*b*) 1,034,289 ft. lb.

(558) In the last example, if the air had been compressed adiabatically to the same pressure, (*a*) what would have been the volume discharged, and (*b*) what work would have been required?

Ans. { (*a*) 52.494 cu. ft.
(*b*) 1,011,317 ft. lb.

(559) (*a*) What is sensible heat? (*b*) What is the specific heat for constant volume?

(560) Change (*a*) 2,917° C., (*b*) 637° C., and (*c*) – 260° C. into their corresponding Fahrenheit temperatures.

Ans. (*a*) 5,282.6°.
(*b*) 1,178.6°.
(*c*) – 436°.

(561) How many units of heat would be required to vaporize 12 pounds of solid mercury having a temperature of – 50°? Use the same value for the specific heat of solid mercury as given in the table for liquid mercury.

Ans. 2,229.595 B. T. U.

(562) What is the weight of 100 cubic feet of hydrogen gas having a temperature of 80° and a tension of 18 pounds per square inch? Ans. .6231 lb.

(563) How many B. T. U. will be required to raise 7 pounds 5 ounces of copper to its melting point from a temperature of 78°? Ans. 1,406.13 B. T. U.

(564) If 2¼ pounds of lead having a temperature of 40° be mixed with 4 pounds of water having a temperature of 65° in a cast iron vessel whose weight is 1¾ pounds and temperature 62°, what will be the temperature of the mixture? Ans. 64.43°.

(565) A wrought iron bar 4 inches square and 20 inches long is heated 1,200°; (*a*) what is its cubical expansion? (*b*) its linear expansion? (*c*) How much larger will the area of a cross-section be?

Ans. (*a*) 7.903 cu. in.
(*b*) .16464 in.
(*c*) .2634 sq. in.

(566) If 10 pounds of nitrogen gas having a tension of 61 pounds per square inch occupy a volume of 73 cubic feet, what is its temperature? Ans. 707.45°.

(567) Why are light clothes cooler than dark ones?

(568) In order to find the temperature of a stove fire, a piece of cast iron weighing 1 pound is placed in it. A copper vessel weighing 1½ pounds is partly filled with 3¼

pounds of water, the temperature of both being 85°. After the cast iron piece has been placed in the vessel of water,

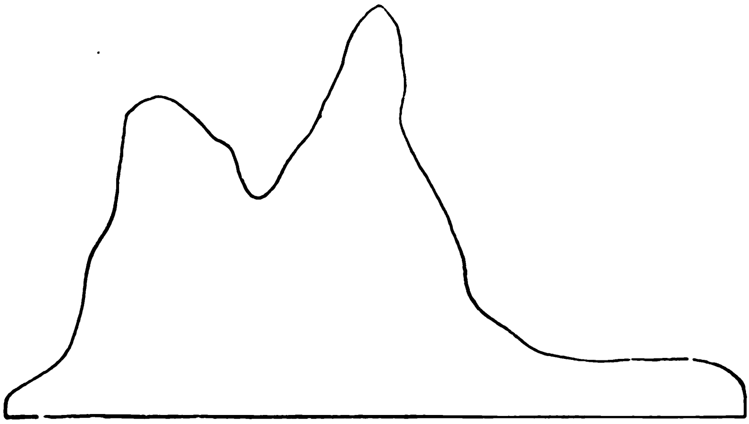

FIG. 3.

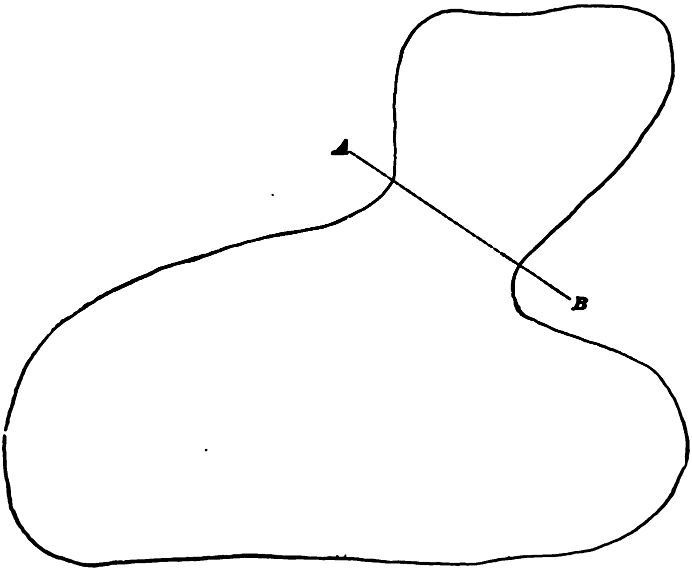

FIG. 4.

the temperature of the mixture is found to be 128°; what was the temperature of the fire? Ans. 1,252°.

(569) Construct a curve similar to the one shown in Fig. 223 by calculating the ordinates. Take the initial pressure as 84 pounds per square inch, and the final pressure as 14 pounds.

(570) Under the conditions of the last example, calculate the work done by the air while expanding.

Ans. 21,673 ft. lb.

(571) What are the absolute temperatures corresponding to (*a*) 96° F.? (*b*) 32° C.? (*c*) 180° C.? (*d*) 650° F.? (*e*) −40° C.?

(572) If 7 pounds of ice at 20° are converted into steam at 212° in 43 minutes, what is the equivalent horsepower of heat energy expended? Ans. 4.971 H. P.

(573) One cubic foot of air at atmospheric pressure is compressed adiabatically 4 times; that is to say, the volume after compression is only $\frac{1}{4}$ that before compression; what work was necessary to compress this air?

Ans. 3,953.28 ft. lb.

(574) Calculate by the method mentioned in connection with Fig. 225 the mean ordinate of Fig. 3, and mark it in the figure.

(575) Calculate in a similar way the area of Fig. 4, using *A B* as the line from which to draw the ordinates.

(576) Four pounds of oil of turpentine at 80° are mixed with a certain quantity of water at 73° in a brass vessel of the same temperature weighing one-half a pound. The temperature of the mixture being 75.61°, what is the weight of the water? Ans. 2.819 lb.

(577) Illustrate your idea of latent heat, specific heat inner work, and outer work.

(578) A cannon ball weighing 120 pounds is fired with a velocity of 1,200 feet per second. If 15% of its kinetic energy on leaving the cannon is converted into heat on striking the target, how many heat units is this equivalent to?

Ans. 517.975 B. T. U.

(579) What do you understand by the terms: (*a*) Hot body? (*b*) Cold body? (*c*) Heat unit?

(580) A tin vessel weighing $1\frac{1}{4}$ pounds contains 30 ounces of water; the temperature of both is 91°. Into this is placed 5 pounds of lead ore having a temperature of 40°. If the temperature of the mixture is 86°, what is the specific heat of the ore? Ans. .0423.

(581) A hot air engine receives air at a temperature of 450°, and exhausts it at 70°; what is its ideal maximum efficiency?

(582) What is the weight of 700 cubic feet of hydrogen having a temperature of 200° and a tension of 20 pounds per square inch?

(583) (*a*) Name the kinds of thermometers in general use. (*b*) Reduce 44° R. to the corresponding Centigrade temperature; (*c*) to the corresponding Fahrenheit temperature.

$$\text{Ans.} \begin{cases} (b) & 55^\circ \text{ C.} \\ (c) & 131^\circ \text{ F.} \end{cases}$$

(584) If a pound of air at atmospheric pressure, and having a temperature of 60°, be compressed adiabatically until its tension is 235 pounds per square inch, what will be its new volume and temperature?

$$\text{Ans.} \begin{cases} 1.8356 \text{ cu. ft.} \\ 704.2^\circ. \end{cases}$$

(585) (*a*) In what three ways may a body be considered to expand? (*b*) To which of these three ways does the expansion of a gas correspond? (*c*) What is absolute temperature?

(586) A hollow copper cylinder is heated by the application of 7,000 B. T. U. If its outside diameter is 10 inches, inside diameter $9\frac{1}{4}$ inches, and length 6 feet, (*a*) what will be its linear expansion? (*b*) Its cubical expansion? (*c*) How much larger will its outside diameter be?

$$\text{Ans.} \begin{cases} (a) & .195 \text{ in.} \\ (b) & 6.63 \text{ cu. in.} \\ (c) & .027 \text{ in.} \end{cases}$$

(587) Twelve cubic feet of gas are heated from 65° to 390°; what is the increase in volume, the pressure remaining constant? Ans. 7.428 cu. ft.

(588) If a piece of cast iron weighing 75 lb. be drawn

back and forth over another piece of cast iron, 2 feet each way, at the rate of 20 times a minute, how many heat units will be developed in one hour, assuming a coefficient of friction of .18? Ans. 83.29 B. T. U.

(589) One-half a pound of air occupies a space of .9 of a cubic foot, and expands adiabatically until its volume is 4½ times as large. If the initial temperature is 150°, (*a*) what is the final temperature? (*b*) The initial pressure? (*c*) The final pressure?

Ans. (*a*) – 130.76°.
(*b*) 125.565 lb. per sq. in.
(*c*) 15.06 lb. per sq. in.

(590) A solid zinc sphere 12 inches in diameter is placed in 8 pounds of boiling water. If the original temperature of the sphere was 70°, what is its increase in volume?

Ans. 1.71 cu. in.

(591) How much would a steel wire rope 900 feet long shorten, if cooled from 90° to 28°? Ans. 4.01 in.

(592) If it takes 5 heat units to raise the temperature of a certain body weighing 26 pounds 1°, what is its specific heat? Ans. .1923.

(593) Draw an isothermal expansion curve by the method shown in Fig. 227, and calculate the work done. Choose your own dimensions, volumes, and pressures.

(594) (*a*) What equivalent work would represent the melting of 13 pounds of sulphur from a temperature of 40°? (*b*) If done in 10 minutes, what would be the equivalent horsepower?

Ans. (*a*) 519,956.5 ft. lb.
(*b*) 1.5756 H. P.

(595) Three pounds of gaseous turpentine at its temperature of vaporization are mixed with 4 pounds of water at 75°; what is the temperature of the mixture? Ans. 203. 11°.

(596) What equivalent work would represent the raising of 25 pounds of lead at 46° to a temperature of 800°?

Ans. 678,353.76 ft. lb.

(597) Eleven and one-half pounds of cast iron at 180°, 42 pounds of brass at 240°, 10 pounds of ice at 10°, and 50 pounds

of water at 120° are mixed together in a lead vessel weighing 20 pounds, and having a temperature of 80°. The ice is added last; what is the temperature of the mixture?
Ans. 91.55°.

(598) If 3 cubic feet of air whose tension is 140 pounds per square inch expand adiabatically to 16 cubic feet, what would be (*a*) the area? (*b*) The mean ordinate? (*c*) The mean pressure of a space corresponding to *A E F L*, in Fig. 229, Art. **1164,** drawn to the same scale? Obtain your result by calculation only.

Ans. (*a*) 25.434 sq. in.
(*b*) 1.9565 in.
(*c*) 39.13 lb. per sq. in.

(599) Change (*a*) – 10° F., (*b*) 25° F., and (*c*) 2,200° F. into the corresponding Centigrade readings.

Ans. (*a*) $-23\frac{1}{3}$° C.
(*b*) $-3\frac{8}{9}$° C.
(*c*) $1{,}204\frac{4}{9}$° C.

(600) Sketch a diagram illustrating a reversible cycle process, and explain in your own language what it means.

(601) What work is necessary to compress 10 cubic feet of air isothermally, from 15 pounds per square inch tension to 85 pounds per square inch? Ans. 37,467.74 ft. lb.

(602) If the indicated horsepower of a steam engine is 520, to how many heat units per hour would this work be equivalent? Ans. 1,323,393.31 B. T. U.

(603) Four pounds of melted zinc at the temperature of fusion are mixed with 10 pounds of water having a temperature of 60°. If the temperature of the mixture is $102\frac{1}{3}$°, what is the latent heat of fusion of the zinc? Neglect the effect of the vessel containing the mixture. Ans. 50.61.

(604) (*a*) In what ways may heat be propagated from one body to another? (*b*) What is meant by convection, and how does it differ from conduction?

(605) Define (*a*) good conductors; (*b*) bad conductors; (*c*) non-conductors.

(606) What is your idea of the reason that radiant heat is transmitted in a vacuum?

(607) (*a*) Change 798 B. T. U. to calories. (*b*) Change 40 calories to B. T. U.

Ans. { (*a*) 201.515 calories.
(*b*) 158.4 B. T. U.

(608) How many foot-pounds of work are equivalent to the heat required to raise the temperature of 7.68 cu. ft. of oxygen gas, having a tension of 18 lb. per sq. in., from 40° to 416° (*a*) when the volume is constant? (*b*) When the pressure is constant?

Ans. { (*a*) 37,379 ft. lb., nearly.
(*b*) 52,429 ft. lb., nearly.

(609) Name the different sources of heat.

(610) (*a*) Explain your idea of isothermal expansion; (*b*) of adiabatic expansion.

(611) If 3.72 cu. ft. of air are compressed to a volume of 1.2 cu. ft., what is the resulting temperature, the original temperature being 68°? Ans. 380°, nearly.

(612) State the first and second laws of thermodynamics.

(613) An air compressor has a stroke of 48″. Assuming the discharge valves to open when the piston has completed 38″ of the stroke, what is the tension and temperature of the air at discharge? Take the original temperature as 40° and assume the compression to be adiabatic.

Ans. { Tension = 134.24 lb. per sq. in
Temperature = 491°, nearly.

STEAM AND STEAM ENGINES.

(ARTS. 1189–1330.)

(614) (*a*) What is saturated steam; superheated steam? (*b*) In what way does saturated steam differ from a perfect gas?

(615) Explain fully the process of changing a quantity of water at 32° F. to steam.

(616) (*a*) Define the following: Total heat; heat of the liquid; latent heat of vaporization. (*b*) How may the "heat of the liquid" be found approximately?

(617) Find by formula **90**, the pressure corresponding to a temperature of 290° F. Ans. 57.92 lb. per sq. in.

(618) Find the total heat of vaporization of 1 pound of steam at a temperature of 318° F.

(619) Explain why saturated steam condenses when expanding in a non-conducting cylinder.

(620) If the pressure of the steam in a boiler is 81 pounds per sq. in. (absolute), what is the volume in cubic feet of 6 pounds of the steam? Ans. 31.7076 cu. ft.

(621) If a pound of steam occupies a volume of 7 cu. ft., what is its absolute pressure? Ans. 60.086 lb. per sq. in.

(622) Answer the following questions by referring to the steam table: (*a*) What is the gauge pressure of steam at 320° F.? (*b*) What is the temperature of saturated steam at a gauge pressure of 110 pounds per sq. in.? (*c*) What is the "heat of the liquid" and "total heat" of a pound of steam at 70 pounds per sq. in. (gauge)? (*d*) What is the

latent heat of vaporization of 3 pounds of steam at a pressure of 67 pounds, absolute?

Ans.
(*a*) 75.18 lb. per sq. in.
(*b*) 343,951°.
(*c*) 286.003 B. T. U.
1,178.266 B. T. U.
(*d*) 2,711.193 B. T. U.

(623) (*a*) What is the isothermal line of saturated steam? (*b*) What curve represents most closely the expansion of saturated steam?

(624) Find from the steam table the weight of 38 cu. ft. of steam at an absolute pressure of 43 pounds per sq. in.

Ans. 3.95466 lb.

(625) Show by a diagram why the expansive use of steam is economical.

(626) How much steam at 60 pounds gauge pressure must be mixed with 300 pounds of water at 55° F. in order that the temperature of the mixture may be 140° F.?

Ans. 23.885 lb.

(627) Find from the table the volume of $4\frac{1}{3}$ pounds of steam at a temperature of 256° F. Ans. 53.226 cu. ft.

(628) How many B. T. U. are required to raise 6 pounds of superheated steam from 310° to 342° F.?

Ans. 92.16 B. T. U.

(629) Calculate by formula **90**, the pressures corresponding to the following temperatures, and compare the results with those given in the steam table: (*a*) 254° F.; (*b*) 377° F.

Ans.
(*a*) 32.094 lb. per sq. in.
(*b*) 190.54 lb. per sq. in.
From the tables:
(*a*) 32 lb. per sq. in.
(*b*) 189.212 lb. per sq. in.

(630) What is the action of steam in a cylinder of conducting material like cast iron? (*b*) What is the use of the steam jackets?

(631) Find by formula **91**, the total heat of vaporization of a pound of steam at the following pressures (absolute), and compare the results obtained with the steam table: (*a*) 210 lb. per sq. in.; (*b*) 88 lb. per sq. in.; (*c*) 37 lb. per sq. in.

Ans. { (*a*) 1,199.596 B. T. U.
(*b*) 1,179.086 B. T. U.
(*c*) 1,161.999 B. T. U.

(632) Show fully why the efficiency of the steam engine may be increased by raising the pressure of the steam used.

(633) Draw a "curve of constant steam weight" by laying off to some scale the volumes taken from column 6 of the table along the horizontal line *O X* and the corresponding pressures as ordinates.

(634) (*a*) Define the following: Head end and crank end of cylinder; forward stroke; return stroke; reciprocating parts. (*b*) When is an engine said to "*run over*"? When does it "*run under*"? (*c*) What is the stroke, and to what is it equal?

(635) (*a*) Define the following: Clearance; real cut-off; apparent cut-off; ratio of expansion. (*b*) The clearance is 7% and the apparent cut-off $\frac{3}{8}$; find the real cut-off and the ratio of expansion.

Ans. (*b*) { Real cut-off, .416.
Ratio of expansion, 2.4.

(636) A non-condensing engine has a clearance of 6%, and cuts off at $\frac{3}{8}$ stroke. If it uses steam at a boiler pressure of 60 pounds per sq. in. (gauge), what is the probable M. E. P.?

Ans. 47.1 lb. per sq. in.

(637) (*a*) What is the I. H. P. of the engine of question 636, assuming the piston to be 11″ in diameter, the length of stroke 18″, and the number of revolutions 210? (*b*) If the mechanical efficiency of the engine is 83%, what is the net H. P.? (*c*) The friction H. P.?

Ans. { (*a*) 85.45 H. P.
(*b*) 70.92 H. P.
(*c*) 14.53 H. P.

(638) (*a*) Define lap; lead; angular advance. (*b*) How does the outside lap influence the "point of cut-off"? (*c*) What is the duty of the inside lap?

at is an indicator diagram? (*b*) What three
are made of them?

The area of an indicator diagram is 3.47 sq. in.
he length of the diagram represents 6 inches of
stroke, the scale of the spring is 40, and the
the engine piston is 16 in. What is the work
per stroke? (*b*) What is the horsepower of
it makes 120 R. P. M.?

ns. { (*a*) 13,953.73 ft.-lb.
{ (*b*) 101.48 H. P.

(641) What are the faults of the diagram shown in Fig. 5?

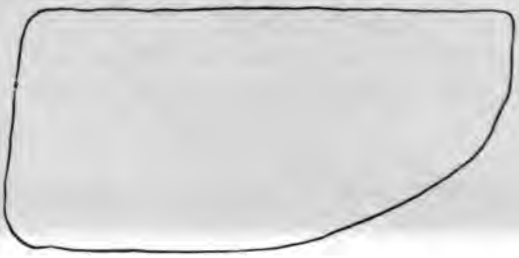

FIG. 5.

How might these faults be partially remedied?

(642) To drive the machinery of a certain shop requires 120 *actual* H. P. Find the probable dimensions of a Corliss condensing engine to do the work, assuming a boiler pressure of 70 pounds (gauge), and a ratio of expansion of 4. Assume a proper piston speed and mechanical efficiency.

(643) (*a*) What is the thermal efficiency of a steam engine using steam at 60 pounds gauge pressure, and exhausting at 2 pounds above the *atmosphere?* (*b*) If the pressure in the above case were raised to 90 pounds (gauge), and a condenser added, giving a back pressure of 3 pounds above *vacuum*, what would be the efficiency?

Ans. { (*a*) 11.55%.
{ (*b*) 23.93%.

(644) (*a*) The piston speed of an engine is 540 feet, the number of revolutions 150; what is the length of stroke?

(*b*) If the length of stroke is 2½ ft. and the piston speed 900 ft. per minute, what is the number of revolutions?

Ans. $\begin{cases} (a)\ 21.6''. \\ (b)\ 180 \text{ R. P. M.} \end{cases}$

(645) Find the water consumption per I. H. P. per hour from the diagram shown in Fig. 6. Scale of spring, 40.

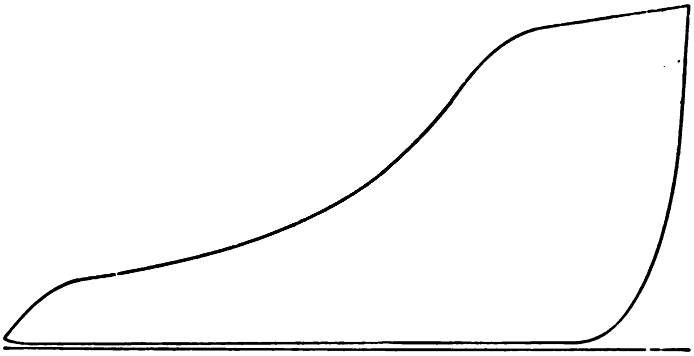

FIG. 6.

SUGGESTION.—Find the M. E. P. by dividing the diagram and measuring the ordinates, and then use formula **101.**

Ans. 22.76 lb. per I. H. P. per hour.

(646) (*a*) Explain fully the thermal advantages of the compound engine; (*b*) the mechanical advantage.

(647) The stroke of a tandem-compound engine is 30 inches. The diameter of the low-pressure cylinder is 32 inches, and of the high-pressure cylinder 19 inches. The M. E. P. of the high-pressure cylinder is 52 pounds, and of the low-pressure cylinder 18 pounds; the number of revolutions, 120. (*a*) Find the I. H. P. of the engine. (*b*) What is the ratio of the work done in each cylinder?

Ans. $\begin{cases} (a)\ 531.29 \text{ H. P.} \\ (b)\ 1.0184. \end{cases}$

(648) (*a*) What defects in the distribution of steam may be shown by the indicator diagram? (*b*) What may be done to remedy the following faults:

1. Admission, release, and compression too early?

2. Admission, release, and compression too late ?
3. Cut-off too early ?
4. Cut-off too late ?

(649) (*a*) Locate the points of cut-off, release, and compression on the diagrams of Fig. 7. (*b*) Assuming them

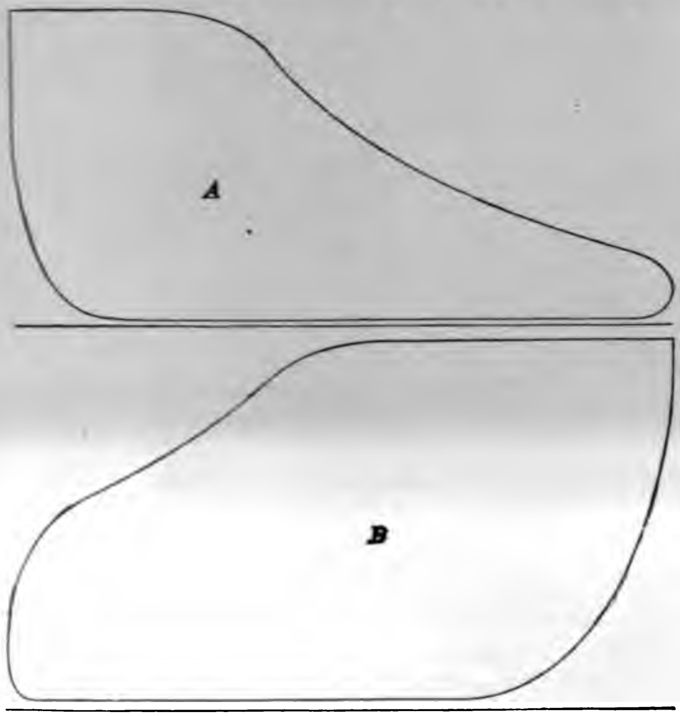

FIG. 7.

to have been taken with a 30 spring, find their average M. E. P.

(650) What should be the condensing surface of a surface condenser for an engine developing 675 I. H. P., and using 26½ pounds of steam per I. H. P. per hour ?

Ans. 1,688.58 sq. ft.

(651) Given, a 30″ × 48″ engine making 80 revolutions per minute. On measuring a crank-effort diagram, it is found

that the fly-wheel is to store up at each stroke 88 foot-pounds of energy per sq. in. of piston area. Assuming a coefficient of unsteadiness of $\frac{1}{60}$, what must be the weight of the fly-wheel rim if the fly-wheel is 20 ft. in diameter?

Ans. 14,251.92 lb.

(652) A compound engine, developing 1,200 I. H. P., has a stroke of 42″, and a ratio of expansion of 5½. The engine makes 70 R. P. M.; the total M. E. P. reduced to the low-pressure cylinder is 42 pounds per sq. in. (*a*) Find the diameter of the low-pressure cylinder. (*b*) Find the diameter of the high-pressure cylinder by formula **103,** and (*c*) by formula **104.**

Ans. { (*a*) 49½ in.
(*b*) 34⅞ in.
(*c*) 32⅝ in.

(653) (*a*) What will be the real cut-off in the high-pressure cylinder of problem 652, if its diameter is found by formula **103?** (*b*) If found by formula **104?**

Ans. { (*a*) .368
(*b*) .426

(654) For what is the Stephenson link used? Describe its action.

(655) The net pressure on the piston of a steam engine is shown by the diagram, Fig. 8. Draw the crank-effort diagram both to a semicircular base and to a straight base. Assume that the length of the connecting-rod is double the stroke.

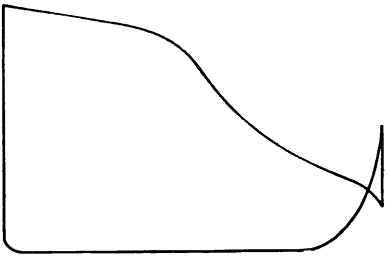

FIG. 8.

(656) The steam enters a jet condenser at a pressure of 4½ pounds above vacuum. The condensing water enters at a temperature of 55° and leaves, after mixing with the exhaust steam, at a temperature of 130°. What weight of condensing water must be used per pound of steam?

Ans. 13.76 lb.

(657) (*a*) Describe the action of the pendulum governor. (*b*) What is the distinction between the duty of a governor and that of a fly-wheel?

(658) A steam engine is to develop 42 H. P. The M. E. P. available is 36.3 pounds per square inch; the engine is to run at 155 revolutions per minute. (*a*) Find the stroke, and (*b*) the diameter of piston, assuming the former to be 1⅓ times the latter. (*c*) What is the piston speed of the engine?

Ans. (*a*) 15 in., nearly.
(*b*) 11¼ in., nearly.
(*c*) 387.5 ft. per min.

(659) (*a*) The diameter of an engine piston is 22"; the pressure at beginning of stroke is 72 pounds per square inch. What is the pressure on the crank-shaft? (*b*) The crank now makes an angle of 60° with the axis of the cylinder, and the steam pressure has fallen to 65 pounds. What is the total pressure on the crank-shaft? (*c*) What is the force tangent to the crank-pin circle?

Ans. (*a*) 27,369.62 lb.
(*b*) 12,354.34 lb.
(*c*) 21,398.25 lb.

(660) The M. E. P. of an engine obtained from the card is 34.6 pounds per square inch. What should be the mean ordinate of the crank-effort diagram expressed in the same units? Ans. 22.03 lb.

(661) A locomotive with 19" × 24" cylinders and 80" drivers uses steam at 150 gauge pressure, and is designed to run at a speed of 60 miles per hour. Under the ordinary load the M. E. P. is 52½ lb. What I. H. P. does the locomotive develop? Ans. 909.72 I. H. P.

(662) The travel of a valve is 5"; the lap is ¾". (*a*) What must be the angular advance of the eccentric? (*b*) What angle will the eccentric make with the axis of the cylinder when cut-off occurs?

Ans. (*a*) 17° 28'.
(*b*) 72° 32'.

(663) The actual horsepower of an engine was experimentally found to be 12.325. The I. H. P. from the diagram was 15.36. (*a*) Find the efficiency of the engine.

(*b*) The engine had a 9″ × 12″ cylinder, and ran at an average speed of 240 rev. What was the M. E. P.?

Ans. { (*a*) 80.24%.
(*b*) 16.6 lb.

(664) Describe the steam-engine mechanism. What office does it fulfil?

(665) Draw the theoretical diagram of a simple steam engine from the following data: Boiler pressure, 70 pounds per sq. in. (gauge); ratio of expansion, 3; back pressure, 2 lb. above atmosphere; clearance, 7%. The steam is compressed to 40 lb. gauge pressure. Assume your own scale of pressures and volumes.

(666) What is a balanced valve, and why are they used?

(667) What is the object of computing the steam consumption from both the point of cut-off and a point near release?

(668) Calculate the size of a simple engine to give 240 *actual* H. P., choosing your own type of engine, a suitable boiler pressure, ratio of expansion, and mechanical efficiency.

(669) (*a*) What should be the scale of the indicator spring used if the boiler pressure is 54 pounds? (*b*) If the boiler pressure is 115 pounds? (*c*) The height of the steam line of a diagram above the atmospheric line is 1.834″. What is the gauge pressure of the steam at admission if the diagram was taken with a 40 spring?

(670) During an engine test 906 pounds of exhaust steam passed through the surface condenser, and 13,580 pounds of water were required for condensation. The average pressure of the exhaust steam was 7 pounds above vacuum. The condensing water entered at an average temperature of 52° F., and left the condenser at an average temperature of 120° F. What was the average temperature of the condensed steam on leaving the condenser? Ans. 148.66°.

(671) The diameters of the high-pressure, intermediate, and low-pressure cylinders of a triple-expansion engine are, respectively, 27″, 42″, and 66″. The M. E. P.'s of the steam

in the three cylinders are, respectively, 72 pounds, 40 pounds, and 16.5 pounds. The stroke of the engine is 4 feet, and the number of revolutions 70. (*a*) Find the I. H. P. of the engine. (*b*) Find the percentage of the work done in each cylinder.

Ans. (*a*) 2,598 I. H. P.
(*b*) 26.9% in high-pressure cylinder.
36.2% in intermediate cylinder.
36.9% in low-pressure cylinder.

(672) What are the distinctive features of the Corliss valve gear? What advantages has it over the plain slide valve?

(673) A Corliss condensing engine cuts off at ¼ stroke, and has a clearance of 2½%. The boiler pressure is 84 pounds. (*a*) Find the probable M. E. P. (*b*) Assuming a piston speed of 500 feet per minute, what must be the diameter of the piston in order that the engine may develop 120 H. P.?

Ans. (*a*) 52.455 lb. per sq. in.
(*b*) 13⅞ in., nearly.

(674) The gauge pressure of the steam in a boiler is 93 pounds. Find its temperature by formula **90.**

Ans. 332.609°.

(675) Draw a skeleton diagram showing the position of the crank and eccentric when the engine runs under, the angle of advance being 37°. The valve and cylinder may be omitted.

(676) Find, by means of formula **93,** the I. H. P. of an 18″ × 24″ engine, whose mean effective pressure is 62.4 pounds per square inch, and which makes 175 revolutions per minute. Ans. 336.825 I. H. P.

(677) In Fig. 9 are shown the indicator diagrams taken from the high and low-pressure cylinders of a tandem compound non-condensing engine. Diameter of high-pressure cylinder, 13″; of low-pressure cylinder, 20″; stroke, 15″; revolutions per minute, 230. Find the horsepower.

Ans. 177.1 H. P.

(678) Take the clearance of both cylinders of the engine in the last example as 10% of the volume of the

high-pressure cylinder, and combine the diagrams into one, as in Fig. 297.

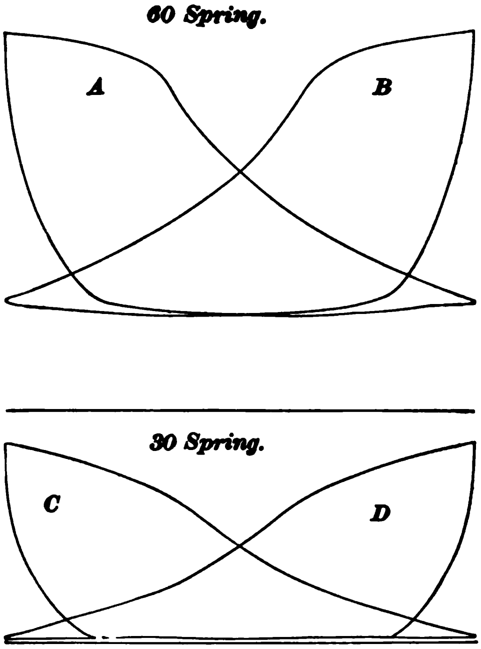

FIG. 9.

(679) Find the water consumption per I. H. P. per hour of a 24″ × 30″ engine making 150 revolutions per minute, and giving a diagram like *A*, Fig. 7. Scale of spring, 30.
Ans. 26.21 lb. per I. H. P. per hour.

(680) (*a*) What is meant by high rotative speed? (*b*) What is the difference between a high rotative speed and a high piston speed? (*c*) Does an engine with a high rotative speed necessarily have a high piston speed?

(681) The pressure on the steam gauge of a boiler is 123 lb.; find the temperature by formula **90**. Ans. 351.49°.

(682) A 300 horsepower engine has a cylinder 22″ in diameter, and a stroke of 18″; that is, it is a 22″ × 18″

engine. When making 200 revolutions per minute, what must be the mean effective pressure?

Ans. 43.406 lb. per sq. in.

(683) In a manner similar to question 675, find the position of the eccentric for an angle of advance of 20°, when the engine has a rocker arm to change the direction of motion of the valve.

(684) In Fig. 10 are shown the diagrams from both ends of the cylinder of a 13″ × 12″ engine running at 300 revolutions per minute. Find the net mean pressure upon the pistons, as was done in Art. **1312,** by combining the expansion line of card *A* with the back-pressure line of card *B*. Scale of spring, 60.

Ans. 36.96 lb. per sq. in.

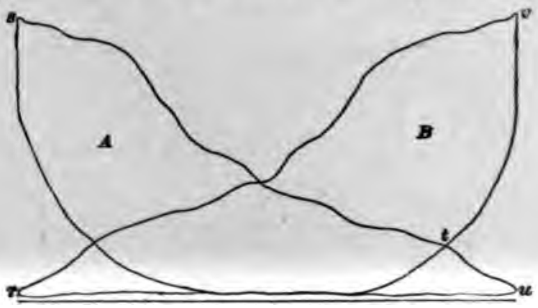

FIG. 10.

(685) Find the indicated horsepower of the engine of example 684.

Ans. 91.66 I. H. P.

(686) Find the water consumption per I. H. P. per hour from the diagrams of Fig. 10, example 684.

Ans. 21.49 lb. per I. H. P. per hour.

(687) Taking the ratio of the length of the connecting-rod to that of the crank as 5 : 1, draw the diagrams of tangential pressures on the crank-pin, both for a circular base and a straight base, of the engine of example 684.

(688) Calculate the weight of the fly-wheel rim of the engine in example 684, taking the coefficient of unsteadiness E as $\frac{1}{50}$, and the ratio n as 5.

Ans. 623 lb.

PRINCIPLES
OF
REFRIGERATION.

(ARTS. 1331–1475.)

(689) What size of pipe would be necessary to carry 320 gallons of brine for a distance of 347 feet with a friction pressure not to exceed 5 pounds? Ans. 5-in. pipe.

(690) State the three methods of supplying the expansion coils with ammonia, and state the advantage of each.

(691) Describe the British Linde circulating system.

(692) In what way does the efficiency of an absorber affect the economy and capacity of the machine?

(693) On what does the economy of an absorption machine depend?

(694) By formulas **115** to **118**, compute the properties of saturated ammonia vapor at a pressure of 14.7 pounds per square inch, absolute.

Ans.
Temperature, – 28.562° F.
Volume of liquid per pound, .02379 cu. ft.
Volume of vapor per pound, 18.018 cu. ft.
Weight of vapor per cubic foot, .0555 lb.
Latent heat, 572.83 B. T. U.

(695) At a temperature of 230° F. and a gauge pressure of 100 pounds per square inch, what is the volume of 1 pound of superheated ammonia vapor? Ans. 3.73 cu. ft.

(696) What is the best test for a slight ammonia leak in the open air, and what for a leak under water?

(697) (*a*) Name the principal objections to the following

fluids as refrigerating agents: Ether; sulphur dioxide; carbon dioxide; ammonia. (*b*) What is the principal advantage of ammonia as a refrigerating fluid?

(698) What precautions must be taken with a sulphur-dioxide machine to insure its life?

(699) (*a*) What is the chief requisite of working fluid for a latent-heat refrigerating-machine? (*b*) What fluids are most available as refrigerating agents?

(700) In the example of Art. **1346**, assume a capacity of 250 pounds of ice per hour and a pressure of 30 pounds gauge in the expansion cylinder at cut-off. Take the other data as given in the example, and make all the calculations that are made in Art. **1346**.

Ans.
Heat abstracted per stroke, 7.925 B. T. U.
Weight of air per stroke, .3393 lb.
Volume of compressor cylinder, 4.276 cu. ft.
Volume of expansion cylinder, 3.435 cu. ft.
Theoretical net horsepower, 5.35 H. P.
Theoretical efficiency, 2.62.

(701) There are three samples of chloride of calcium, *A*, *B*, and *C*, each weighing 1 pound and dissolved in half a gallon of water. On testing with a hydrometer, the *A* sample shows 1.165; the *B* sample, 1.173; and the *C* sample, 1.157 Which is the best grade of calcium?

(702) What are the relative merits of a vertical and horizontal generator or retort?

(703) In changing from salt brine to chloride-of-calcium brine, what must be done with the salt brine?

(704) Give a brief statement of the relative advantages of the direct-expansion and brine systems.

(705) A certain plant uses per minute 247 gallons of brine, which is cooled 4½° by the refrigerating-machine. What is the tonnage of the plant? Ans. 44.46 tons.

(706) How many gallons of brine per minute would a brine-pump have to discharge in a 40-ton plant with a temperature range of 6½°? Ans. 153.85 gal.

(707) Which is more desirable for brine-circulation work, return bends with close centers or with wide centers?

(708) Wherein does the operation of an ammonia absorption machine differ from that of a compression machine?

(709) (*a*) A warehouse has four rooms. Room *A* requires 29 tons; room *B*, 12 tons; room *C*, 18 tons; and room *D*, 4 tons. The temperature of the inlet brine is 5° and of the return 9°. How much brine does each room require? (*b*) If room *A* is 60 feet from the engine room; room *B*, 30 feet from *A*; room *C*, 75 feet from *B*; and room *D*, 40 feet from *C*, what size of brine mains would have to be run under the given conditions, the friction head between any two rooms not to exceed 12 feet?

Ans. (*b*)
Pipe to *A*, 4 in.
" " *B*, 3 in.
" " *C*, 3 in.
" " *D*, 1¼ in.

(710) Why are air-cocks put on brine coils? Where should they be placed?

(711) Make all the calculations given in the problem of Art. **1374,** using the following data:

Ice-melting capacity,	12 tons.
Temperature in condenser,	85° F.
Temperature in refrigerating coil,	20° F.
Initial and final temperatures of cooling water,	60° and 80° F.
Compressor, single-acting,	70 strokes per minute.

Ans.		
	Theoretical volume of cylinder,	.415 cu. ft.
	Theoretical horsepower,	8.48 H.P.
	Cooling water per minute,	136.81 lb.
	Theoretical efficiency,	6.6

(712) Explain fully how the suction pressure influences the capacity and efficiency of the ammonia compression machine.

(713) How does the condensing pressure influence the efficiency of a compression refrigerating-machine?

(714) State the advantages and disadvantages (*a*) of single and double acting compressors; (*b*) of vertical and horizontal compressors. (*c*) What advantage has the direct-acting machine over the indirect-acting machine?

(715) (*a*) What expedients have been devised to reduce the superheating of ammonia during compression? (*b*) Explain how wet compression cools the compressor cylinder.

(716) A double-acting compressor having a cylinder displacement of 357 cubic feet per minute is running under the following conditions:

Head pressure, 180 pounds per sq. in., gauge
Back pressure, 10 " " " " "

(*a*) What is the theoretical tonnage? (*b*) Deducting 30 per cent. for loss due to superheating, what is the capacity?

Ans. { (*a*) 75.254 tons.
(*b*) 52.68 tons.

(717) Explain what you understand to be the difference between standard and superheated ammonia vapor.

(718) To produce a certain ice-melting effect, the volume of the compression cylinder of an ammonia machine is 2 cubic feet. For the same capacity under the same conditions, about what should be the volume of the compression cylinder (*a*) of a sulphur-dioxide machine? (*b*) of an ether machine? (*c*) of a carbon-dioxide machine?

Ans. { (*a*) 5⅝ cu. ft.
(*b*) 34 cu. ft.
(*c*) ½ cu. ft.

(719) (*a*) In what way is the cycle of the ammonia compression machine incomplete? (*b*) In what way is it proposed to make the cycle complete in the case of carbonic-acid machines?

(720) Describe fully the action of the ammonia compression machine, illustrating your description by sketches.

(721) (*a*) Explain the production of cold by the adiabatic expansion of a gas. (*b*) Why is there no fall of temperature if the gas expands adiabatically without performing work?

(722) An air machine has an expansion cylinder with 12-inch bore and 15-inch stroke and makes 100 displacements per minute. The air temperature after expansion is — 43°; the temperature of the cold room is 30°; and the pressure is that of the atmosphere. What is the theoretical capacity of the machine? Ans. .817 ton.

(723) State the principal causes that tend to decrease the efficiency of an air machine. Which of these causes, in your opinion, is the most serious, and why?

(724) In what way is the commercial efficiency or economy of a refrigerating-machine measured?

(725) (*a*) Give three expressions for the theoretical efficiency of a refrigerating-machine, and show how they are obtained. (*b*) In what way do the temperatures of the hot body and cold body influence this efficiency?

(726) (*a*) What is the capacity of a refrigerating-machine? (*b*) What is the unit of refrigerating or ice-melting capacity? (*c*) What is meant by ice-making capacity, and how does this differ from ice-melting capacity?

(727) If it is desired to have a brine temperature of 10° below zero F., what salt should be used, and what should be the minimum gravity of the solution?

(728) What are the relative merits of sodium and calcium chloride for making brine?

(729) Explain fully the difference between the direct-expansion system and the brine system of refrigeration. Illustrate the description by sketches.

(730) (*a*) Explain the principle of action of the vacuum refrigerating-machine. (*b*) For what purpose is sulphuric acid used in a vacuum machine? (*c*) What are the disadvantages of vacuum refrigerating apparatus?

(731) Name the four kinds of absorbers, and state the advantages of each.

(732) State the advantages and disadvantages of the carbon-dioxide refrigerating-machine.

(733) How can the quality of the anhydrous ammonia in an absorption machine be determined without drawing out a sample?

(734) Describe the action of the ammonia absorption machine. Illustrate your description with sketches.

(735) Of what advantage is the use of the exchanger of the absorption apparatus?

(736) A refrigerating-machine abstracts 635,342 B. T. U. in 6 hours and 20 minutes. What is the refrigerating capacity of this machine?

(737) Explain how cold is produced by evaporation of liquids. What class of refrigerating-machines operate on this principle?

(738) Explain the operation of a refrigerating-machine working through a Carnot cycle (*a*) using a perfect gas, as air, for a working fluid; (*b*) using a saturated vapor for a working fluid.

(739) In what respect does the actual cycle of a refrigerating-machine differ from the ideal Carnot cycle?

(740) Show how the second law of thermodynamics applies to the refrigeration process.

(741) An ammonia meter being placed in the main feed line of a direct-expansion system, it is found that the plant requires 113 cubic feet of anhydrous ammonia liquid in 2 hours and 12 minutes, in order to keep the rooms cold. The head pressure of the ammonia is 135 pounds, and the suction pressure is 25 pounds, gauge. What should be the capacity of this plant in tons? Ans. 83.09 tons.

(742) A plant has a brine-pump with a brine end 6 inches in diameter and a measured stroke of 10½ inches. The pump runs at 82 displacements per minute. The brine that it circulates is cooled from 15° to 11°. The brine has a specific gravity of 1.2, at which the specific heat is .75. What is the capacity of the machine after deducting 20% for the slip of the brine-pump? Ans. 12.78 tons.

(743) (*a*) How may the approximate commercial efficiency of a compression machine be obtained ? (*b*) By what process may the commercial efficiency be determined with accuracy ?

(744) Explain fully how to determine the quantity of heat rejected to the condenser during a test of a refrigerating-machine.

(745) (*a*) Explain the method of determining the capacity of a refrigerating-machine when the direct-expansion system is used. (*b*) Explain how the capacity may be determined when the brine system is used.

(746) (*a*) What are the principal data required in a test of a refrigerating-machine? (*b*) What is the primary object of such a test?

(747) What is meant by a *heat balance?* Of what use is it in a test of a refrigerating plant? Explain how the heat balance is formed in the case of a compression machine, and also in the case of an absorption machine.

(748) In one of Prof. Denton's tests, the following data were observed:

Average head pressure	**161 pounds, gauge.**
Average suction pressure	**27.5 pounds, gauge.**
Temperature brine inlet	**36.36° F.**
Temperature brine outlet	**28.45° F.**
Brine circulated per minute	**2,374 lb.**
Initial temperature of condensing water	**54° F.**
Final temperature of condensing water	**82.8° F.**
Water circulated per minute in condenser	**601.5 lb.**
Water circulated per minute in jacket	**14 lb.**
Initial and final temperatures of jacket water	**54.3° and 83.45°.**
Weight of ammonia circulated per minute	**28.32 lb.**
Specific gravity of brine	**1.174**
Specific heat of brine	**.783**
Average horsepower of steam cylinder	**88.63**
Average horsepower of ammonia cylinder	**71.2**

From these data compute (*a*) the heat per minute taken up by the brine; (*b*) the heat given to the condenser per minute; (*c*) the heat given to the water-jacket per minute.

(*d*) Form the heat balance. (*e*) Calculate the capacity in tons per 24 hours. (*f*) Assume 3 pounds of coal per horsepower per hour and compute the commercial efficiency, that is, the pounds of ice melted per pound of coal.

Ans. (*a*) 14,703.5 B. T. U.
(*b*) 17,359.3 B. T. U.
(*c*) 408.1 B. T. U.
(*e*) 67.6 tons.
(*f*) 23.26 lb.

REFRIGERATING
AND
ICE-MAKING MACHINERY.

(ARTS. 1476–1600.)

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(749) Describe the process of charging an absorption system with aqua ammonia. Illustrate your description by a sketch showing essential parts of the system.

(750) What is the indication of too heavy feeding or expanding, and what effect does this have on the packing of a compressor?

(751) How many tons of refrigeration are required to cool 327 barrels of beer to 40°, the wort being cooled by the upper coil of the Baudelot cooler to 68°; the specific gravity of the wort is 1.06, and the average specific heat is .892? The cooling is to take place in 4 hours. Ans. 47.76 tons.

(752) Describe the wet-plate system of ice making. What is the objection to this system?

(753) (*a*) Explain why the ammonia pump should be set below the absorber receiver. (*b*) What gauges are required in an absorption plant, and where are they located?

(754) (*a*) For what reason should the foundation of an ammonia condenser be level? (*b*) What is the advantage of a deep tank for a submerged condenser? Explain fully. (*c*) Describe the construction, setting, and insulation of the brine tank. Illustrate by sketches.

(755) Give the three methods employed in cooling and ventilating cold-storage rooms. State the advantages and disadvantages of each.

(756) (*a*) Name some miscellaneous uses of refrigeration. (*b*) Why is the direct-expansion system not applicable to hotel refrigeration?

(757) (*a*) How many feet of 2-inch direct-expansion pipe are required to cool a chill room having a capacity of 15,000 cubic feet? (*b*) How many feet of 1¼-inch pipe should be provided for the circulation of the brine, the latter being at 15°?

Ans. { (*a*) 1,154 ft.
(*b*) 3,000 ft.

(758) (*a*) What advantage has the absorption system over the compression system in ice making? (*b*) By what is the capacity of an ice plant governed? (*c*) How long a time is required to determine correctly the capacity of a plant?

(759) A packing house is to have 10,000 cubic feet of chill-room space, 15,000 cubic feet of pickling, and 50,000 cubic feet of storage room; 120 hogs and 50 beeves are killed daily, the average weight of the hogs being 240 pounds and that of the beeves 700 pounds. (*a*) How many tons of refrigeration will be required? (*b*) How many lineal feet of 2-inch direct-expansion pipe for each class of rooms will be necessary?

Ans. { (*a*) 22 tons.
(*b*) { 770 ft. for chill room.
1,625 ft. for storage and pickling room.

(760) Make a sketch showing the arrangement of the distilled-water system in an absorption ice-making plant.

(761) Describe the forced-air circulation system of cooling and ventilating cold-storage rooms. Make sketches showing essential features of the system.

(762) If the condensing pressure of an absorption machine is 135 pounds and the steam pressure 60 pounds, what is the lowest specific gravity to which the liquor in the still can be reduced?

(763) What is the object of the vacuum test of a compression plant? Describe the test.

(764) (*a*) What is meant by the term *tank surface?* (*b*) What methods are employed to agitate the brine in the

tank? Which method is used when cold storage is combined with ice making?

(765) Describe the course of the distilled water in a compression ice plant, can system. Make a sketch showing the parts necessary to the description.

(766) State the points to be observed in the storage of (*a*) eggs; (*b*) dairy products; (*c*) fruits.

(767) (*a*) Describe the process of cooling wort. (*b*) About how many barrels of wort may be cooled per ton of capacity under ordinary conditions?

(768) (*a*) To what tests should the compression machine be subjected? (*b*) Describe the test of the water and steam pipes.

(769) The contents of an ice can is 2.25 cubic feet. What is the net weight of the cake made?

(770) Describe the indirect-radiation method of cooling cold-storage rooms, using sketches to illustrate the description.

(771) What system of chilling should be employed in the refrigeration of packing houses? Give reasons.

(772) A person owns an abbatoir having 8,000 cubic feet of chill-room space, 20,000 cubic feet of pickling space, and 60,000 cubic feet of storage space. The capacity of his refrigerating-machine is 25 tons. How many hogs can he chill in his chill room if the average weight of a hog is 250 pounds? Assume in this case that 1 ton of refrigeration will cool 8,000 cubic feet of space. Ans. 444 hogs.

(773) What is the best method of insulating cold-storage rooms that are liable to be damp, like those of a packing house or a brewery?

(774) (*a*) Describe the pressure test of a compression machine, illustrating the description by sketches. What is the object of this test? (*b*) In this test what precautions must be taken to prevent the formation of explosive gases?

(775) How many tons of refrigeration are required for

a brewery where 100 barrels are brewed daily, the capacity of the storage cellars being 60,000 cubic feet?

Ans. 14 tons, nearly.

(776) Describe briefly the essentials of the can system and plate system of ice making. State the advantages and disadvantages of each system.

(777) (*a*) Describe the gas forecooler; use sketches to illustrate the description. (*b*) What materials are used in the water-filters?

(778) (*a*) Why must the water used in ice making be distilled? (*b*) From what source is the distilled water usually obtained? (*c*) What is the ordinary yield of a can ice plant per pound of coal?

(779) (*a*) Describe the operation of shutting down the compression machine. (*b*) Describe the starting of the machine.

(780) Describe two methods of insulating exposed pipes or surfaces.

(781) What is the capacity of an absorption machine making anhydrous ammonia, if the ammonia pump has a bore of 3⅛ inches in diameter, a stroke of 10 inches, and makes 60 strokes a minute? Ans. 40 tons.

(782) (*a*) What precaution is necessary to make an air-space an effective insulator? (*b*) Name several filling materials and state the advantages or disadvantages of each.

(783) Make a sketch showing (*a*) the insulation of an outside brick wall; (*b*) the insulation of an upper floor.

(784) (*a*) How may a leak in the generator coil be detected? (*b*) How may a leak in the exchanger coil be detected?

(785) (*a*) What are the conditions to be observed in the preservation of articles in cold storage? (*b*) How is the cold-storage temperature maintained? (*c*) What is the objection to placing freezing rooms above cold-storage rooms?

(786) (*a*) Describe the course of the condensing water in

an absorption plant, using necessary sketches. (*b*) What variation of the usual arrangement is made in very warm climates?

(787) Describe the process of charging a compression plant with ammonia. Illustrate your answer by a sketch showing only the parts of the plant and the valves and fittings that are essential to the description.

(788) The specific gravity of wort of a certain strength is 1.053 at 60° F. What will be its specific gravity (*a*) at 39° F.? (*b*) at 73° F.?

Ans. $\begin{cases} (a) \ 1.0562 \\ (b) \ 1.0511 \end{cases}$

(789) (*a*) What are the causes of a too high head pressure? (*b*) How is air removed from the condenser?

(790) (*a*) What limits the capacity of an absorption machine? (*b*) What should be the steam pressure used with horizontal generators? (*c*) What precaution must be observed with vertical generators? (*d*) What is the maximum quantity of aqua ammonia that should be circulated per ton of capacity in an absorption machine?

(791) (*a*) Describe the steam piping of a compression refrigeration plant. (*b*) Name the essential valves and cocks required on a compression plant.

(792) (*a*) What is the object of using a weak-liquor cooler? (*b*) How much water is required by an absorption plant under ordinary conditions?

(793) (*a*) Describe the process of charging an absorption system with anhydrous ammonia. Use sketches. (*b*) How may air be expelled from an absorption plant?

(794) (*a*) What precautions must be taken in handling anhydrous ammonia? (*b*) How may the ammonia charge be removed from a compression machine?

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